S·A·E JOURNAL

PUBLISHED BY THE SOCIETY OF AUTOMOTIVE ENGINEERS, INC.

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Vol. 34 APRIL, 1934 No. 4

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Publication Office, 56th and Chestnut Sts., Philadelphia, Pa.; Editorial and Advertising Departments at the headquarters of the Society, 29 West 39th St., New York, N. Y. Western Advertising Office, Room 2-136 General Motors Bldg., Detroit, Mich.

Highway Accidents *Are* Avoidable

Wide World Photo

By R. A. L. Bogan

Vice-President, The Greyhound Corp.

MORE than 33,000 persons are killed and a million persons are injured each year because of the abuses of the highway by vehicle operators. A large proportion of the accidents now resulting in fatality and serious injury can be avoided; but, if accidents increase in the same proportion as they have during the past four years, the fatality toll undoubtedly will mount in a very short time to at least 50,000 lives annually.

There are a large number of causes for the rising toll of accidents on the highway, but the worst causes of accidents, in the opinion of the writer based upon many years of interesting experience in accident prevention, are as follows:

1. Street and road intersections.
2. Passing other vehicles going in the same direction, resulting in collisions with vehicles coming in the opposite direction.
3. Vehicles running off the highway because they have been crowded by another vehicle coming in the opposite direction.
4. Running into the rear of the car ahead going in the same direction.

While the writer does not intend to imply that the above are the only types of accidents which result in serious injury or fatality, they do nevertheless cover in a broad sense practically all such accidents.

Accidents resulting from collisions at intersections can be prevented in a large degree by (a) greater alertness on the part of the driver, (b) reduced speed if view is curtailed, (c) not taking safety for granted, (d) the driver taking full responsibility for the protection of his life and his vehicle from another person's reckless driving.

Accidents resulting from passing other vehicles going in the same direction and resulting in collisions with vehicles coming in the opposite direction can be prevented in a large degree if the driver of the vehicle about to overtake the one ahead will remember that the safe place is on the right-hand side of the center line of the roadway, in which position he should stay until he is certain he can pass with safety.

Accidents resulting from vehicles running off the highway because they have been crowded by other vehicles coming in the opposite direction can be prevented only when drivers are properly educated in the courteous use of the highway; by that I mean when the driver believes sufficiently in safety to protect the rights of others.

Accidents resulting from running into the rear of the car

[This paper was presented at the Annual Meeting of the Society in Detroit, January, 1934.]

ahead can be prevented in a large degree if the driver will keep far enough behind the car ahead for safe braking distance.

Greyhound has had many years of experience with this great problem of highway safety, and we believe that the causes of most of the difficulties are well enough known, so that effective cures are possible in a large percentage of cases. We believe there is a pressing need for drivers' license laws in all of the United States. Most of the unqualified drivers now using the highway are those who are driving their own vehicles. I make this statement, not to discredit the careful and safe private driver, but to point out that the real cause of accidents in the United States today is the result of unsafe practices on the part of individual drivers who while they are driving unsafely think they are perfectly competent to drive safely. A drivers' license law, if properly administered, will eliminate much of the difficulty.

Fewer Accidents in Licensing States

In connection with the above, it is interesting to note that, in the states where drivers' license laws are now in effect and where the state police are active in administering this law, there were 31 per cent fewer automobile accidents since the passage of drivers' license laws than would have occurred if the deaths in those states had continued at the same rate as in the states where they had no such laws.

The automobile, because it is cheap, fast and economical to operate, has caused most Americans to want to be in a hurry. In the right hands the automobile has been a boon to our national life; but in the wrong hands it has become a weapon of destruction. There are about 26 million automobiles in the United States. Many of the drivers of these cars know little or nothing about their operation, and therefore are a real menace on the highway. Highway accidents involve some definite factors. They are the vehicle, the highway upon which they operate and the driver. The proper design of the automobile and the construction of the highway will contribute largely to the safety factor in the operation of automobiles.

Increase of speed of automobiles demands increased safeguards in the car and on the highway, as well as increased ability and care on the part of the driver.

Compulsory inspection of brakes and headlights ought to be a requirement in all states, but the real danger on the highway is neither the road nor the car, it is the driver.

Motor buses and trucks are safer than private automobiles. The chief reason for this is that the owners of fleets have given greater consideration to the education and discipline of drivers and to the establishment of safety rules for the guidance of the drivers. These owners have thus created in the minds of the men a definite interest in safety. Control of vehicles in hazardous places has been given real consideration. Railroad tracks, curves and slippery roads have been acknowledged by these owners and fleet drivers as being dangerous places; and, since the realization of this danger has been definitely embedded in the minds of the drivers, a greater respect for safety has been implanted in their minds.

The National Safety Council keeps a close record of highway accidents. During a recent four-year period they advised that fatalities involving privately-owned automobiles have increased 50 per cent and during the same period fatalities involving all commercial vehicles have decreased 12 per cent. After a one-year period ending last June all inter-city buses, according to their report, had 70 per cent fewer accidents per 100,000 miles of operation than all other types of motor vehicles combined. The safety of motor buses was proved by the National Safety Council after it conducted a nation-wide contest with all types of fleet owners.

The proof of the possibility of operating safely on the highways is best shown by the following figures which are taken from a report of the National Safety Council for December, 1933, showing a no-accident record of commercial vehicle fleets. These records are statistically accurate:

Greyhound Management Co. (Detroit Division). Inter-city bus operation. 985,942 vehicle-miles. Terminated Sept. 8, 1933.

Gulf Companies (Ft. Worth Division). Petroleum (trucks and passenger cars). 886,717 vehicle-miles. Terminated May 4, 1933.

Oklahoma Gas and Electric Co. (Central Division, Sapulpa, Okla.). Public Utilities (trucks and passenger cars) 835,295 vehicle-miles. Terminated Oct. 17, 1932.

South Shore Laundry, Chicago Ill. Laundries. 810,750 vehicle-miles. Unbroken Nov. 15, 1933.

Pocahontas Transportation Co., Bluefield, W. Va. Inter-city bus operation. 779,159 vehicle-miles. Unbroken, Sept. 30, 1933.

You will note that the Greyhound companies operating in this area, right in the city of Detroit, where traffic is heavy and driving is necessarily difficult because of the thickly-populated area, have established a world's record of 985,942 vehicle-miles without an accident in a period which terminated on Sept. 8, 1933. This is attributable in a large degree to the interest in safety of Manfred Burleigh, who is the vice-president in charge of operations in this area. The other outstanding thing about the records described above is that the large fleet owners have shown such an interest in safety as to effect such excellent records. Drivers of privately-owned automobiles have had no such training and therefore have never considered the safety angle in the same degree as the commercial driver.

License Law Emphasizes Safety

A properly-designed drivers' license law will inculcate in every driver's mind the importance of safety and courtesy on the highway. The protection of the rights of all users of the highway by state police or other police officials is the next move.

The careless or reckless bus driver or truck driver is disciplined by his boss, but the private owner has no discipline unless it is by education through a drivers' license law or by the police. Discipline has a wholesome effect on all drivers.

Marcus Dow, whom I consider one of the genuine authorities on highway safety and highway safety methods, has frequently said that "safety education means more than any other effort in the development of a careful driver." He has repeatedly stated that the slogans "Safety First" or "Be Careful" are excellent slogans, but they are not sufficient to make for safety on the highways. The campaign of education must be thorough. It also must be interesting and instructive. Speed may be dangerous at 50 m.p.h., and it may also be dangerous at 10 m.p.h., but speed is always dangerous when not properly controlled. The slow-going vehicle that fails to keep up with the traffic may be and generally is more dangerous than the fast-moving vehicle. We should record all accidents and judge the causes and effects. Accidents should be enumerated and classified for public benefit, and they should also be publicized.

Chronicle *and* Comment

By
Norman G. Shidle

BACK to Saranac!

June 17-22 are the dates of the 1934 Summer Meeting which returns to Saranac Inn, Lake Saranac, N. Y., for the first time since 1929.

The July, 1929, issue of the S.A.E. JOURNAL records in lively action pictures and interesting text the story of the Society's former visit to the northern New York resort. "The place was ideal," says the report, "the accommodations and the service generally satisfactory, and the program varied without being too extensive."

Now in prospect, as in retrospect, Saranac holds wide attractions as a summer meeting site. And this year, as in 1929, it appears that the program will be "varied without being too extensive."

The Passenger-Car Activity is planning to follow through on the technical discussion of noise problems which attracted so much attention at the Annual Meeting this year, while further practical data on streamlining and engine developments will furnish the basis for other sessions sponsored by this activity. Truck and transport subjects, fuel discussions, aviation topics and a variety of other questions are being scheduled as well. So far have plans already advanced, in fact, that the May issue of the JOURNAL probably will be able to carry a somewhat detailed outline of what is planned.

With hotel rates lower than at any regular summer meeting in many years, the 1934 Saranac party bids fair to set new records of enjoyment and value.

TOTAL enrollment mounted to 135 for the series of six lectures on lubricants sponsored by the Cleveland Section at the Case School of Applied Science. The series ended on Feb. 27. A bibliography covering the sources of the lecture material and abstract of the discussion were made up in mimeographed form and distributed to those who took the course.

So successful were these lectures as a method of treating a subject of special interest to S.A.E. members that the Course Committee, headed by Hoy Stevens, is considering possible subjects which might be handled in similar manner next winter.

WHILE the motoring public has been admiring the striking new features incorporated in the 1934 passenger car models, manufacturing executives back at the factories, in devising efficient methods of doing new jobs, have been matching the ingenuity and progressiveness of the design engineers. Day after day they are proving fallacious the idea current a few years ago that production efficiency had about reached its limit.

A particularly pertinent view of current manufacturing advances reached us the other day from a well-known S.A.E. production executive who recently returned to active service in the industry after an absence of a little more than a year. "One's general perspective of the industry," he writes, "is materially broadened by a rest such as I have had, which brings a better appreciation of the continual progress, determination and resourcefulness associated with the men in it."

Then he goes on to say: "Engineering developments for 1934 again have necessitated a decided re-vamping of manufacturing methods. Materials have changed from one type to another in many parts of the car. Automotive suppliers are equally affected and kept on their toes to provide new products to replace those obsoleted. Indeed, it continues to be a game of ***unfinished work ahead***. . . .

"Recognition is due to automotive plant owners for the courage they have displayed in continuing engineering developments during the hazardous three years just passed.

"The NRA has injected other interesting problems into management and, I am frank to say, many of us are taking a post-graduate course."

T*WENTY Years Ago This Month:* Seventy leading automotive engineers extol the value of S.A.E. standards in signed statements to S.A.E. BULLETIN. "Adherence to the Standards of the Society has saved our own company thousands of dollars yearly in lowered material costs and in lessened production complications," writes Howard E. Coffin of Hudson. Other confirming opinions come from W. R. Strickland of Peerless; W. G. Wall of National; C. S. Crawford of Cole; B. B. Bachman of Autocar; C. B. Whittelsey of Hartford Rubber Works; H. W. Alden of Timken-Detroit; S. O. White of Warner Gear; J. G. Vincent of Packard. . . .

Arrangements are announced for the Summer Meeting of the Society to be held at Cape May, N. J.; also the tentative itinerary of the Second European Visit of the Society. . . .

Recent changes of position include that of F. E. Moskovics, from secretary, Jones Electric Starter Co., to Nordyke & Marmon in Indianapolis Ralph R. Teetor, from mechanical engineer, Light Inspection Car Co., to a similar position with the Teetor-Hartley Motor Co.; and Charles E. Duryea from Duryea Motor Co. of Saginaw to Cresson-Morris Co., Philadelphia.

Acute Automotive Radio Problems Are Yielding

AN automobile radio receiver is similar in many respects to a "home" type radio,' but has certain requirements that differ considerably. Fundamentally it is a device to convert into sound energy, radio signals which are selected at the will of the driver or operator. The sound energy must be of ample intensity to be heard throughout the car at any speed.

The automobile receiver then is analogous to the "home" receiver in that it takes the signal energy at radio frequencies from an antenna, amplifies and converts it into energy at audio frequency, again amplifies, and feeds it to the loud speaker. The loud speaker is in turn a device to convert electrical energy into sound energy as is desired.

Starting with the radio signal we find that, due to the antenna limitations in cars, the automobile receivers must in general be considerably more sensitive than home-type receivers to give satisfactory operation under driving conditions of varying signals. The selectivity must be comparable to that of an average home receiver, although there is also the danger of too great selectivity causing difficulty in tuning while driving.

The maximum audio power output of an automobile receiver is generally lower than that of better home-type receivers, although in many cases the output level used is greater with the car outfit due to masking effect of driving noises.

The loud speakers used in cars are of the same types as used in home receivers, although, as discussed below, they have become much smaller than those considered satisfactory in a good home receiver. In the one-piece car receivers, the speakers in most cases are of the same size and limited capabilities as in the sub-midgets.

The frequency characteristics of automobile receivers are in general poorer than good home receivers (leaving out of consideration the smaller midgets), although the balance of high and low-frequency responses must be carefully controlled so that the result in the car is pleasing. High-fidelity receivers have no place in the automotive field at present, as they would not show up any better than other receivers under the operating conditions and would cost considerably more.

The power input of an automobile receiver is a factor which the designer has to watch much more closely than does the designer of a home receiver. With the home receiver the use of 25 or even 50 per cent more power from the electric light system to secure minor desirable operating characteristics is not a serious thing and is never noticed by the owner. The same percentage increase in power consumption from the car battery over what is required for satisfactory characteristics may be very disagreeable to the car owner.

This paper was presented at the Annual Meeting of the Society, Detroit, January, 1934.

FOLLOWING a lively review of the constructive cooperative activity between automotive and radio engineers which is resulting in solution of mutually pressing problems, Mr. Graham, in this article, urges among other things:

Standardization of radio mounting space;

Universal use of the S.A.E. standard on grounded polarity of the car battery;

Use of good antenna as being "the cheapest and best form of amplification, considering customer satisfaction";

Larger batteries and generator capacities on radio-equipped cars;

Elimination of ignition interference to receivers in other services.

He emphasizes strongly this last topic, pointing out that "with the advent and popularity of so-called all-wave receivers, this problem is brought acutely to the attention of the public. . . ." The automotive industry, he feels, will be eager to avoid popular condemnation in this respect.

Compactness with good operating characteristics is essential to the car radio, whereas many home radios are designed with a large chassis just to look large even though the actual amount of apparatus does not require such size. In home-type receivers a large baffle area provided by the cabinet is possible and desirable to enable the reproduction of the fundamental low frequencies. Compactness should not be carried to extremes at the cost of operating satisfaction as discussed below.

Car receivers must also be designed to have a minimum pickup of interference from the car electrical system.

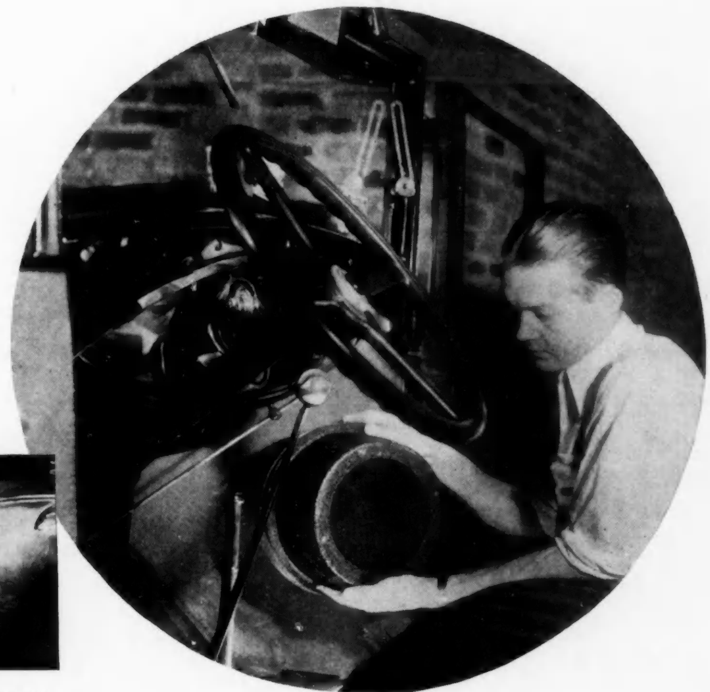
Obviously, the design of a small receiver for car service with the above mentioned requirements is a considerably greater problem than the design of a home receiver, and for that reason the recommendations of the radio engineer as to size (within reason), tubes, power supply, etc., should be heeded.

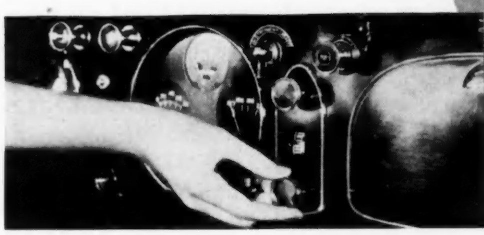
Several years ago the first radio receivers for use in automobiles appeared on the market, after a period of experimental

to United Action

By Virgil M. Graham

Radio Engineer, Stromberg-Carlson Telephone Mfg. Co.





Courtesy *Radio Retailing*

activity of mounting home-type battery-operated receivers in cars. The first automotive radios were in general large and cumbersome and required large "B" batteries for their operation. The tubes used were those designed for home-type radio equipment, and generally were of the filament type, fragile and not wholly satisfactory for the purpose. The audio power output of these receivers was low because of the limitations of the "B" power supply and of the tubes.

The next step in the development of automotive radio was the use of motor-generators or dynamotors of various types for the "B" power supply. Then, came the introduction by the tube companies of "heater type" tubes with a 6.3 volt rating for use directly across the car battery. These tubes had a comparatively low current drain for the heaters and were constructed in smaller bulbs. This was a big step in advance, as such tubes gave the receiver engineer means of doing a better job in circuit designs and in compactness without sacrifice of performance than had been possible previously. More recently there has been made available complex tubes, each performing more than one function in the receiver, making possible a reduction in the number of tubes employed as well as still greater compactness. Now the dynamotor began to be replaced by the less expensive vibrator type of power supply, either using a rectifier tube or being of the self rectifying type.

About that time, the radio manufacturers felt that an effort to standardize on mounting space in cars for the radio equipment was desirable. With this problem in mind, along with several others concerning ignition interference, antennas, etc., the Radio Manufacturers Association in the Spring of 1932 approached the Society of Automotive Engineers with a suggestion that a joint committee on automotive radio be formed. The S.A.E. very kindly agreed to the organization of such a committee, and it was suggested that the R.M.A. prepare definite recommendations to present to the first meeting with the automotive engineers. This was done and on Sept. 16, 1932, the first meeting of the joint committee was held. A very cordial cooperative effort was thus started and several joint meetings have been held since that time.

One of the proposals presented to the joint committee set down mounting space requirements for the receiver chassis and loud speaker. These were never finally adopted since the design of automotive radio equipment was changing rapidly, mainly on account of economic factors which were pushing the cheaper "one-piece" receiver into use. Such receivers, with extremely small speakers, fewer tubes, fewer tuned circuits, etc., were admittedly inferior in performance to the larger two-piece and three-piece equipment, but found considerable favor due to lower initial cost.

It is unfortunate that some of the automotive people should feel that the radio manufacturers in making proposals for mounting space standardization were attempting to ask the car manufacturers to "design their cars around the receivers." Such is not the case, as what the radio manufacturers desire is cooperation with the car manufacturers in giving the public better automotive radio, an objective in which both groups must of necessity be interested. It is obvious that without some such standardization the radio manufacturers must build special receivers for many makes of cars. Of course, such a procedure is not economical. The ideal of economical standardization and public satisfaction would be attained if such essential mounting space and dimensions and electrical connections were standardized, so that each automotive radio manufacturer could build a receiver which would fit in any make of car, be easily installed, and operate properly. While not impossible, this ideal has not been attained, but it can be by the continuation of concentrated cooperation between the automotive engineers and the radio engineers. Each group must recognize the problems of the other and work together toward their solution and the ultimate result of better automotive radio. Naturally, along with this aim goes the necessity that the equipment must be so designed and priced that

each group can make a reasonable profit in manufacturing, handling, and servicing.

It is not the purpose of this paper to propose any definite standards recommendations, but there is one item of which every radio manufacturer would urge mention. This is the great need of uniform use by the car manufacturers of the S.A.E. Standard on the grounded polarity of the car battery. Lack of uniform practice necessitates incorporation of pole-changing provisions in the receivers, and causes a very considerable number of avoidable service complaints. The adoption of uniform practice in this matter will obviously be a great help to the automotive radio industry.

The car antenna presents another problem which demands joint efforts of the automotive and radio engineers. The automotive man has the structural details and cost to concern him, while the radio man has the need of sufficient pick-up of signal to give satisfactory operation. It is admitted that the better signal pick-up of a better antenna gives more desirable results than the addition of sensitivity to a receiver on a poorer antenna. Therefore, the radio engineer would present the point to the car manufacturer that a good antenna is the cheapest and the best form of amplification, considering customer satisfaction.

Squeezing Apparatus Not Beneficial

The desire to squeeze the radio down to as small a space as possible has not been beneficial to performance as mentioned above, and the radio engineer would again point out that while the cost may be slightly greater and the space requirement larger, the use of a good-sized speaker will give better satisfaction both from the standpoint of tone quality and the better utilization of the electrical power output of the audio system of the receiver. In this regard it is interesting to refer back to the original R.M.A. "Recommendations" dated July 28, 1932, where the following item is found:

II Mounting Space of Radio Equipment.

B. Speaker mounting space.

1. Bulkhead mounting—9 inches by 9 inches minimum. (The radio manufacturers feel that this is the minimum space required for a speaker that will give customer satisfaction.)

At the first meeting of the joint S.A.E-R.M.A. committee on automotive radio there was considerable discussion of the current consumption of automobile receivers. The answer to this problem is rather completely involved with the operating characteristics of the receiver such as audio power output and sensitivity. The audio power output is probably the largest factor in determining the total battery drain of the receiver, as any audio system so far devised is only moderately efficient and requires filament and plate power to give output power. Therefore, when a receiver with high output power is desired, the user must be willing to supply filament and plate power which in turn comes from the car battery. If the greater and greater power output is to be desired from automobile receivers, it is agreed that batteries and generators of greater capacities must be supplied at least on radio equipped cars. Some receiver designs are more efficient than others as regards battery drain for the same operating characteristics, but it seems to be generally admitted that greater capacities in batteries and generators would be very desirable with present designs of receivers.

Possibly the voltage-regulated type of generator will come into greater use as a result of increased use of radio. At the Oct. 20 meeting of the joint committee, J. T. Fitzsimmons of Delco-Remy stated that the tendency in automotive generating systems would be toward slightly higher voltages with less variation in voltage. He said that this was the result of the tendency to use larger generators with higher outputs, so regulated that the output will be in proportion to the load demands on the system. Again the radio engineer would urge that this trend be followed by all car manufacturers.

The automotive engineers have accomplished wonderful advancements in making the cars more comfortable and quieter to increase the pleasure and ease of riding. The radio engineer would feel that he was not giving good counsel if he failed to urge the use of better operating characteristics and features of automobile radio so as to be in keeping with the car improvements.

Every one knows the importance of proper suppression of interference from the ignition system of cars in which radio receivers are installed. There is at present a joint research subcommittee of S.A.E. and R.M.A. working on this problem under the chairmanship of L. F. Curtis of the United American Bosch Co. This subcommittee is working on the problem of simplification of the means of suppressing this interference, and it is expected that their findings will be of very considerable value to both industries.

There is another problem of ignition interference, which is in some ways considerably more serious than that just mentioned. This problem is that caused by interference to receivers in other services. With the advent and popularity of so-called all-wave receivers, this problem is brought acutely to the attention of the public.

The higher frequency bands are coming into more and more extensive use by the public. This means that there will be a popular demand for the clearing up of the interference situation just as quickly as the users of such radio equipment find out the cause. The demands for the public utility companies to clear up interference in the broadcast band are good examples of what will happen if the situation is not cleared before the clamor arises. In the case of interference with other services, the introduction of the ultra-high frequencies for police work, aircraft landing beacons, and allied uses, the demand for elimination of interference will also be pressing. Both of these services are coming widely into use. To suggest a possible extreme case of effect of ignition interference, consider a transport plane fog-bound and unable to follow the landing beam because of a car or bus parked at the airport with the motor running.

Public May Seek Action

While the real demand for action will undoubtedly come from the public whose entertainment is interrupted, the responsibility for interference with the other services will be laid on the automotive people.

An interesting comment has been made that whenever a radio service has had a frequency assignment in or near the broadcast band that sooner or later the operators have moved out of that assignment as a result of pressure of public sentiment. Notable examples are the amateurs and some ship services, the former having an adjacent band and the latter a few coincidental frequencies with broadcasting. The public does not care where the interference is originating, but it does demand the elimination.

The second outstanding point is that the regular interference suppression means used when a receiver is installed in the car is effective in the higher frequency ranges. In other

words, if all cars, buses, and trucks had suppression means applied there would be no complaint. The automotive people apparently do not feel that the use of resistor suppressors in the high tension circuits is the final desirable answer to the problem. This is indicated by the following paragraph written by Ray Ellis of the United Motors Service, Inc., on Dec. 15, 1933:

"We would much prefer to approach the problem from a different angle, that is, one of suppressing the interference inside the hood of the car perhaps without the use of the suppressors. We seem to have been successful in doing this on the cars using broadcast receivers, and we believe that this could be worked out all right. There is no question but that suppressors in the ignition circuit on some cars will cause a decrease in performance, and if you increase the voltage on the coil to offset this to some extent, you again build up radiating fields which increase your disturbances."

The public will not care what means is taken to suppress this interference as long as that means is effective.

While the radio people are at present taking the burden of such complaints, a considerable portion of that burden will be shifted to the automotive industry when the public realizes the situation. Of course, some portion of the total interference will be laid to the public utilities for interference from signs, dial telephones, trolley "noise", etc., but the automobile ignition interference is generally readily recognizable by the user of a receiver and can be traced more easily than some of the other interferences.

The radio industry is naturally anxious to clear up obstacles to sales, and it is felt that the automotive industry will be anxious to avoid any popular condemnation in this respect.

To let the imagination work a little it is not difficult to foresee a radio enthusiast sitting at his receiver in the evening and making comments as follows:

"There come the Smiths in their Blank car. They are about six blocks down now and you can hear the car plainly, as that make is one of the worst disturbers."

"Our next door neighbors just came in. They have an 'X' car and it hardly interferes at all, as the makers have partially suppressed the noise in it."

"Well the Smiths are getting closer now, and we had better turn the set off until they get into the garage."

And about that time the station announcer gives the call letters which, of course, are lost.

This may sound foolish, but unless action is taken by the car manufacturers promptly such situations will arise.

In Canada the Radio Division of the Department of Marine has the power to consider that any sources of radio interference, such as signs, power lines, or automobiles, are unlicensed transmitters and on that basis demand the elimination of the interference. It is not inconceivable that the same power might be given to the Radio Commission in the United States if the demand of public opinion becomes strong.

In the home electrical appliance field there is a situation to which the automotive condition may well become analogous. Within the last few years the manufacturers of such appliances, vacuum cleaners, mixing machines, hair dryers, etc., have been forced by the public to design their devices to prevent radio interference. Now, more and more purchasers of such equipment are asking to be sure that it is non-interfering.

Meetings Calendar

S.A.E. Summer Meeting

Saranac Inn, Saranac Lake, New York, June 17-22, 1934.

Chicago—April 3

Bal Tabarin, Hotel Sherman; dinner 6.30 P.M.

The Hesselman Engine, Its Design and Application—A. W. Pope, Jr., research engineer, Waukesha Motor Co.

Butane versus Gasoline as a Transport Fuel —D. P. Barnard IV, assistant director of research, Standard Oil Co. of Indiana.

Cleveland—April 16

Cleveland Club; dinner 6:30 P.M.

Technical Education—George W. Smith, Jr., vice-president in charge of production, White Motor Co.

Dayton—April 12

Engineers Club; dinner 6:30 P.M.

The Development of the Electric Starter—William A. Chryst, consulting engineer, Delco Products Corp.

Leisure Moments in the Engineer's Life—Dr. R. G. Stott, dinner speaker.

Detroit—April 16

Book-Cadillac Hotel; dinner 6.30 P.M.

Indiana—April 19

The Athenaeum, Indianapolis; dinner 6.30 P.M.

Metropolitan—April 19

The Roger Smith, New York City; dinner 6:30 P.M.

Subject—Trends and Future Developments in Truck Design—Joseph Geschelin.

Milwaukee—April 18 and 19

Hotel Pfister

The Section will cooperate in the National Tractor and Industrial Power Equipment Meeting of the Society.

New England—April 10

Walker Memorial, Massachusetts Institute of Technology, Cambridge; dinner 6:30 P.M.

Recent Diesel Engine Development—Myron S. Huckle, president, U. S. Diesel Corp.

Moving pictures of the 1933 Summer Cruise of the Section will be shown.

Northern California—April 10

San Francisco; dinner 6.30 P.M.

Subject—Lubrication.

Northwest—April 27

Engineers Club, Arctic Bldg., Seattle; dinner 6:30 P.M.

Philadelphia—April 18

Inquirer Bldg.; dinner 6:30 P.M.

Railcar Development—C. O. Guernsey, chief engineer, J. G. Brill Co.

Pittsburgh—April 19

Auditorium, Pittsburgh Chamber of Commerce; joint luncheon meeting of the Pittsburgh Section and the Pittsburgh Chamber of Commerce.

Importance of the Automobile to the Return of Prosperity to Pittsburgh's Diversified Industries—C. E. Wilson, vice-president, General Motors Corp.

St. Louis—April 19

Coronado Hotel.

The Place of the Truck in Modern Transportation Engineering—G. R. Gwynne, transportation engineer, White Co.

Southern California—April 19

Cafeteria, Richfield Oil Bldg., Los Angeles; dinner 6:30 P.M.

Washington—April 11

Corcoran Hall, George Washington University, Washington, D. C.

Mind or Micrometer—C. B. Veal, research manager, S.A.E.

Human Engineering—P. R. Wheeler, Navy Department.

News of the Society and

National Tractor Meeting Will Speed Vital Energizing of Widespread Group

NINE years of progress in the tractor and industrial equipment field will be caught up and crystallized at a National Tractor and Industrial Power Equipment Meeting to be held in Milwaukee, April 18 and 19. The meeting is being sponsored and arranged by the recently revived National Tractor and Industrial Power Equipment Committee of the Society.

C. G. Krieger, agricultural engineer, Ethyl Gasoline Corp., is chairman of the Committee.

Fowler McCormick, International Harvestor Co., leads a list of speakers who will provide a comprehensive review of the most important problems facing the agricultural and industrial power fields. The Milwaukee meeting will be the first of the kind which has been held by the Society since one in Chicago nine years ago. It will be the first opportunity in the history of the Society that petroleum technologists and agricultural and industrial power engineers have had to meet in open forum for direct consideration of the problems which involve both vocations.

The number of papers appearing at each session has been limited purposely in order to allow the very representative list of engineers and executives who will attend the meeting adequate time for complete discussion of the questions to be raised.

In the nine years which have passed since the last National Tractor meeting many problems have arisen for which no satisfactory solution has been found. Tractors are being equipped with rubber tires, storage batteries, self-starters, headlamps and other equipment involving the desirability of standardization. For instance, 121 sizes of rubber tires are being used on tractors and farm machinery. It has been estimated that the number of sizes required could possibly be reduced to 12 by concerted action on the part of interested manufacturers. In addition to authoritative statements on the relation of agricultural economics to farm equipment requirements there will be detailed engineering surveys on the requirements of tractor and industrial engines and the fuel situation relative to them. A paper by A. C. Staley, Chrysler Corp., will summarize the results of a questionnaire sent to oil companies and tractor and industrial engineers.

The complete program of the meeting follows:

National Tractor and Industrial Power Equipment Meeting

Hotel Pfister, Milwaukee, Wisconsin
April 18 and 19, 1934

WEDNESDAY, APRIL 18

MORNING SESSION

Topic—The Relation of Engineering to Manufacturing and Merchandising in the Farm Machinery Industry
Speaker—Fowler McCormick, International Harvester Co.
Session Chairman—P. W. Eells, LeRoi Motor Co.

AFTERNOON SESSION

Topic—Some Diesel Tractor Problems
Speaker—H. H. Howard, Caterpillar Tractor Co.
Topic—Spark Ignition Engines for Agricultural and Industrial Use
Speaker—E. R. Jacoby, Continental Motors Corp.
Session Chairman—J. B. Fisher, Waukesha Motor Co.

(Continued on page 32)

Council Presents Changes Proposed in Constitution

AT the Business Session of the Annual Meeting, J. H. Hunt, then chairman of the Constitution Committee, submitted, at the request of the Council, proposals for changes in paragraphs C20 and C21 of the Constitution. On March 14, these proposals were studied by the Council and are presented herewith for the information of the membership. Further action on the suggested changes will take place at the Summer Meeting, to be followed by a letter ballot covering the voting membership of the Society.

The proposals are that:

C20 be changed to read as follows:

"The Council may, at its discretion, by a three-fourths ballot vote of its duly elected members, change any then existing schedule of initiation fees for membership in the various grades. No increase in fees shall be effective until after two months' notice by publication in the JOURNAL of the Society, or by letter to the membership."

C21 to be changed to read as follows:

"The Council may, at its discretion, by a three-fourths ballot vote of its duly elected members, change any then existing schedule of dues for membership in the various grades, and may provide for the payment of dues for periods of three, six or twelve months. No increase in dues shall be effective unless announced by publication in the JOURNAL of the Society, or by letter to the membership, at least one month prior to the beginning of the period in which the increase is to be first effective."

By-Law Addition Voted

At the Session on March 14, 1934, the Council voted its final approval of the addition of the following By-Law:

B-32a—The Secretary and General Manager, after his appointment at the first Council meeting after the Annual Meeting of the Society, may, with the approval of the Council, appoint an Assistant General Manager of the Society.

Patents Policy Set

From time to time the Council has received suggestions that the Society should patent devices developed as a result of Committee activities. At the March 14 meeting of the Council a statement of policy on patents, submitted by J. H. Hunt for the Patents Committee, of which B. B. Bachman is chairman, was adopted. The statement reads as follows:

"Patents are assets only to organizations which are able to capitalize their

the Council

possible value by pushing the commercial application of the devices covered. The Society of Automotive Engineers cannot do such work effectively. Furthermore, the determination of a patent position by prosecuting a patent application involves the risk of becoming involved in very expensive interference proceedings and dealing with prior rights which cannot be foreseen under the American system of patent procedure. The Patents Committee believes that it is unwise to use the funds of the Society in the attempt to secure patent rights. The Committee therefore recommends that the Council adopt the general policy that patents will not be applied for on behalf of the Society. In case some special reason should arise in which it may be desirable to restrict the activities of others, a careful survey of the situation should be made with the idea of securing any needed protection for the Society by other means than by patent application."

Members of the Patents Committee in addition to Messrs. Bachman and Hunt are G. C. Arvedson, National Automobile Chamber of Commerce; and M. W. McConkey, patent attorney. The statement adopted by the Council was submitted as representing the unanimous sentiment of the Patents Committee.

Publications Policy Studied

The Council approved also a report of J. H. Hunt, chairman of the Publication Committee, outlining procedure on publications matters in general and on selection of material for the JOURNAL in particular. The report supplements a previous review made by the 1933 Publication Committee of which G. W. Lewis was chairman and completes the reformulation of publication policies begun last year.

The current report establishes a more definite procedure for selection of papers for the JOURNAL by committees of competent readers in the various activities, who review papers falling within their technical province and rate them confidentially to a scale which has been determined. The 1934 Publication Committee consists of J. H. Hunt, chairman, G. W. Lewis, G. L. McCain, P. C. Ritchie, and S. W. Sparrow.

E. F. Lowe Made Assistant General Manager

Edward F. Lowe — John A. C. Warner — C. B. Whittelsey, Jr.

JOHN A. C. WARNER, secretary and general manager, announces the appointment of Edward F. Lowe as assistant general manager and C. B. Whittelsey, Jr., as assistant secretary of the Society.

Mr. Lowe has gained a widespread reputation in automotive affairs through twenty years of active association with the industry in engineering and executive capacities. Mr. Whittelsey advances to his new position from the post of assistant to the general manager in which he has served so successfully since 1926. Prior to this, Mr. Whittelsey had been engaged in engineering and research work with U. S. Rubber Co., Curtiss Engineering Corp. and Stone & Webster, Inc. He was graduated from Yale Sheffield Scientific School in 1921.

Born in Chattanooga, Tenn., Mr. Lowe first came in touch with the automotive industry by way of the paint and varnish business. He was at one time manager of the Toronto plant of Brandram-Henderson Co., makers of paints and varnishes for the automobile and furniture industries.

In 1914, he went to Detroit as one of the organizers of the Monarch Governor Co. and pioneered development of the automatic type of governor in the motor truck field. Later he sold his interest in the Monarch concern and became interested in K.P. Products Co. of N. Y. of which he became general manager. After K.P. Products merged with Handy Governor Corp. of Detroit, Mr. Lowe became vice-president in charge of sales of the latter organization.

A full member of the Society for ten years, Mr. Lowe has been consistently active in developing various phases of its work. After serving as treasurer and vice-chairman of the Metropolitan Section in successive years, he became chairman of that Section in 1928. In 1929 he was chairman of the general membership committee and in 1932 was a member of the Transportation and Maintenance Activity Committee, representing that division of the Society on the general membership committee.

Since 1931 he has been a member of the Parts and Fittings Division of the Standards Committee and last year served as chairman of the House Committee. At the time of his appointment as assistant general manager he was a member of the Detroit Section.

An unusually wide acquaintance throughout the industry, constant contact with its technical thought over a long period and practical executive experience in business combine to give Mr. Lowe unusual qualifications for the important post for which he has been chosen.

Behind the Scenes With

Knock Ratings

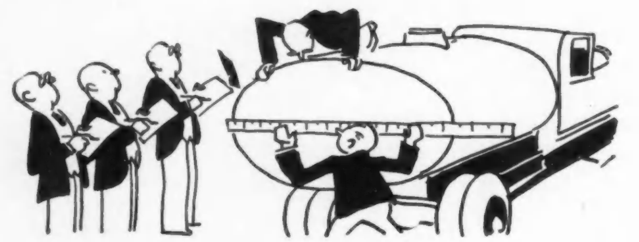
STUDY of the correlation problem between road and laboratory knock ratings during 1934 was recommended by the Subcommittee on Methods of Measuring Detonation to the Cooperative Fuel Research Committee and a tentative proposed program was outlined and submitted to the Steering Committee at its Feb. 6 meeting. The program in general principle was approved at that meeting and its details discussed at length. Copies are being circulated to the members of the Committee for further consideration and comment in writing prior to March 1.

Effort is being made to secure full cooperation from interested foreign organizations and the tentative program will be subject to such modifications as may be necessary to suit these special foreign interests and requirements.

Truck Tank Standardization

THE joint committee on standardization of truck tanks, representing the American Petroleum Institute, the Tank Truck Manufacturers and a subcommittee of the Motorcoach and Motor Truck Standards Division sent out on March 1 a letter which summarizes its work along the following lines.

At a meeting in New York at which the groups were represented all information available and relevant to the problem was collected and tentative suggestions were made regarding the oval diameters for tanks. This information has been submitted for possible final agreements by the three groups concerned.



The study of the general project is to be continued with respect to tank lengths. This also involves the question of the present S.A.E. Standards on CA dimensions and also consideration of trends in truck design such as installation of the engine over the forward axle and cab location in front of the present conventional position.

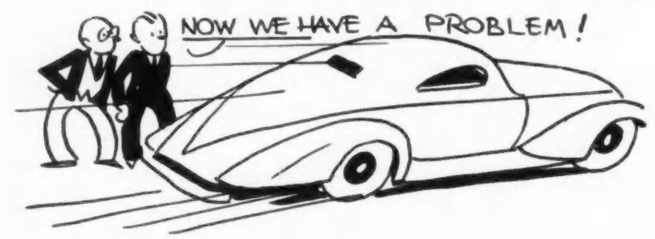
Chassis Lubricants

THE lubricants division of the Standards Committee has decided to discontinue publication of the present general information on free-wheeling transmission lubricants viscosity numbers (pp. 436-438 S.A.E. HANDBOOK 1933 edition). A. L. Clayden, Sun Oil Co. has been appointed chairman of a subdivision to revise the present S.A.E. recommended practice on transmission and rear-axle lubricant viscosity numbers particularly with reference to providing for Winter transmission lubricants for low temperature operating conditions.

Proposed classifications for chassis lubricants have been submitted to grease manufacturers and automotive manufacturers for criticism and comment. This information is to be used as a basis for definite recommendations by the chassis Lubricants Sub-Division to the Lubricants Division at the 1934 Summer Meeting.

Crankcase Oil Stability

THE special Subcommittee appointed last October brought in a report to the Crankcase Oil Stability Research Subcommittee at its meeting on Jan. 25 at which J. B. Fisher, the newly-appointed chairman, presided. The report was approved and the special Subcommittee's request



for an extended period in which to obtain and analyze data was granted.

Various methods of test were described at the meeting with considerable discussion but it was the general consensus of opinion that there is no entirely satisfactory or acceptable test at the present time.

It was further agreed that with the higher speeds and the wider use of the solid-injection engine, sludging is becoming a problem of increasing importance and therefore the Committee should continue its endeavor to solve the problem by providing a satisfactory test method.

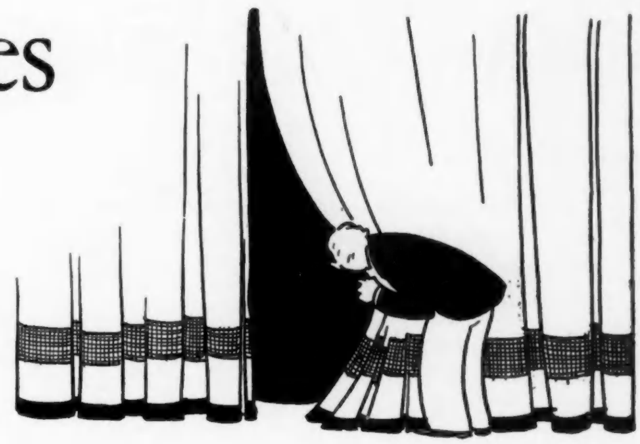
Extreme-Pressure Lubricants

THE Extreme-Pressure Lubricants Subcommittee at its meeting on Jan. 24 made provision for the construction of a lubricants test machine of the design developed in the course of the research, and for securing estimates of the cost of building a limited quantity of these research machines to be used by the laboratories cooperating in the project.

The committee will continue to conduct tests on load-carrying capacity, on samples submitted by individual companies, at a price to cover the expense of the testing at the Bureau and allow a return to the extreme-pressure lubricants fund.

The newly appointed Chairman, W. S. James of the Studebaker Corp., and the members of the Executive Committee, have been requested by the Committee as a whole to outline a program for the year 1934, the amount of the budget required and a proposal for future financing and to transmit this material to the Committee by letter.

the Committees



Wrench-Head Bolts and Nuts

MEMBERS of the Screw Threads Division of the Standards Committee have received for consideration a communication from the Sectional Committee on Bolt, Nut and Rivet Proportions relative to international unification of width across flats and the designation of the bolt length and thread length. Subcommittee No. 2 of the Sectional Committee is collecting opinions and recommendations relative to these standards for use in determining its action in making a tentative recommendation to the Technical Committee of the International Standards Association, which is studying this project.

Front Wheel Alignment

PASSENGER-CAR 1933 wheel alignment specifications corrected to show whether caster is measured with or without load were collected by the Front Wheel Alignment Subcommittee and sent to the leading technical automotive journals for their information and for correction wherever necessary of their published alignment specifications. The Committee is now collecting 1934 specifications.

Representatives of commercial vehicle makers were added to the Committee forming a commercial vehicle division, and enlarging committee influence to the truck and bus field. Specifications on trucks and buses will be included in the Committee's 1934 compilation of specifications.

A Proposed Code of Basic Instructions for Wheel Alignment was formulated and discussed at the Committee's January meeting. The purpose of such a code is to make available a basic set of instructions for aligning front ends, which will enumerate at least the essential elements, serve as a guide of practice for state motor vehicle commissioners, and clarify the question of: "What are the requirements for front end inspection?" The proposed code is being further revised.

Highways Research

TURNING direction indicators have been the subject of study by the Highways Subcommittee during the past few months. An effort is being made to secure data concerning the use of indicators in England and Germany but progress of the Committee to date is not sufficient to warrant a report.

Cooperative Fuel Research

THE Cooperative Fuel-Research Steering Committee at its meeting on Feb. 6 approved the conduct of a series of cooperative road tests at Uniontown this Summer as part of a program of further study on the problem of correlation between road and laboratory ratings by the motor method (A.S.T.M. Designation D 357-33T).

Special invitations to send representatives and, if possible, cars for participation in the tests are being extended to interested groups in Europe and Canada.

Representatives of the Institution of Petroleum Technologists in England, which has adopted as standard the C.F.R. Engine and a slight modification of the motor method, have expressed a desire to participate in the road tests; and a communication of recent data from France indicates that a number of private and public organizations there are working toward some agreement and possible standardization of the method. It is hoped, therefore, that the forthcoming tests will be participated in by these and other foreign groups.

Felts

WORK is getting under way rapidly on revision of specifications for felts used in the automotive industries. A committee with G. M. Wolf, General Motors, as chairman working in conjunction with Committee D-13 of the American Society for Testing Materials is drafting standards specifications for methods of testing felts. Subsequent S. A. E. specifications will probably incorporate these by reference except where it is necessary to develop additional production specifications or additional test methods for the peculiar requirements of the automotive industry.

Automobile Storage Batteries

TENTATIVE revisions on the present S.A.E. Standards for automobile storage batteries (pp. 100-103 S.A.E. HANDBOOK 1933 edition) were drafted by members of a conference on storage batteries which was held during the Annual Meeting in Detroit. The specifications apply only to lead acid storage batteries for automotive purposes, the proposed revisions being under the following heads: ratings, life test, location of battery parts, type designations and markings, and terminal posts. Revised tables for dimensions of passenger-car batteries, small passenger-car batteries, motor-truck batteries and motorcoach batteries have been sent out for tentative approval by the conference and the Electrical Equipment Division.

Safety Glass

ORGANIZATION work is rapidly being completed for a sectional committee on safety code and methods of tests for safety glass. The committee is being organized under American Standards Association procedure. S.A.E. representatives on the committee will be W. H. Graves, chief metallurgist, Packard Motor Car Co. and H. B. Haskins, director of patent and new devices section, Fisher Body Corp.

Papers Available in Mimeographed Form

UNTIL current supplies are exhausted, copies of the papers listed are available in mimeographed form at a cost of 25 cents per copy to members; and at 50 cents per copy to non-members.

Orders must be accompanied by remittance and should be addressed to Sessions Secretary, Society of Automotive Engineers, 29 West 39th St., New York, N. Y.

Almen, J. O., and Wilson, E. E.
Analysis of Intake Silencer Problems

Barnard, D. P.; Barnard, E. R.; Rogers, T. H.; Shoemaker, B. E.; and Wilkin, R. E.
Causes and Effects of Sludge Formation in Motor Oils

Bleicher, C. E.
External Broaching

Boelter, L., and Rusk, D. O.
The Relation of the Mechanical Construction of Headlamps to Their Performance Upon the Roadway

Brettell, Clinton
How Economies in Motor Vehicle Operation Can Be Effected from an Operator's Standpoint

Briggs, Commander W., and Fox, M. L.
Body Noise

Broulhiet, Georges
Quality Objectives for Engineers—An official communication from the Societe des Ingenieurs de l'Automobile de France

Brown, W. C., and Roper, V. J.
The Well-Lighted Car

Bull, A. W.
Tire Noise

Chandler, F. F.
Notes on Steering

DeSmet, E. C.
Planography—The New Science of Surface Design

Drake, H. W.
Problems of the Fleet Operator

Falge, R. N.
Modern Headlighting Requirements

Fisher, J. B.
Combustion Problems on Automotive Diesel Engines

Fisher, J. B., and Bower, L. L.
Trends in Tractor Engine Design

Fitzsimmons, J. T.
Problems and Tendencies in Electrical Equipment

Fodor, Nicholas
Hydraulics of High-Speed Fuel Injection

Foley, Hamilton
Manufacture and Magnetic Inspection of Hollow-Steel Propellers

Frye, Jack
Aircraft Maintenance on Scheduled Service

Gagg, R. F., and Farrar, E. V.
Prediction of Altitude Performance of Aircraft Engines with Gear-Driven Superchargers

Haarz, W. G., Jr.
Beauty Sells Cars in 1934

Hazard, S., Jr.
Sound Absorption and Deadening

Horner, F. C.
Looking Forward in the Field of Transportation

Hunsaker, J. C.
Airships for Commercial Purposes

Jacoby, E. R.
Practical Design Consideration of the Internal Combustion Engine Structure

Johnson, W. M.
A Résumé—and Some Conclusions

Kuttner, Julius, and Rippere, J. B.
Ignition Delay of Diesel Fuels Measured by Bouncing Pin in C.F.R. Engine

Lansing, R. P.
Starters for Diesel Engines

Lichty, L. C., and Carson, G. B.
Engine Friction Analysis

Macauley, J. B.
Fuel Economy from the Engine Designer's Point of View

Marble, J. C.
The Lysholm-Smith Hydraulic Torque Converter

Moss, F. A.
Air Conditioning and Relative Refinements for Auto Bodies

Norris, R. F.
The Automobile Motor Considered as a Sound Source

Nutt, Arthur
Detonation Rating of Aviation Fuels

Orr, J. M.
Predetermined Operating Requirements for Purchasing Equipment

Peterson, C. D.
Multi-Range Transmissions

Prescott, F. L.
High-Output Poppet-Valve Cylinders

Prudden, T. M.
Noise Treatment in the Automobile

Rendel, T. B.
European Automotive Diesels

Robertson, D. D.
Hydraulic Action in Piston Ring Design

Rothrock, A. M.
A Photographic Study of Combustion in a High-Speed Compression-Ignition Engine

Shepard, E. H.
The Economy Fallacy

Smith, C. W.
Comparative Tests of Pneumatic Tires and Steel Wheels on Farm Tractors in Agricultural Operations

Stewart, J. P., and Risk, T. H.
Factors Affecting Oil Consumption

Teetor, R. R.
Conformity of Cylinders, Pistons and Rings

Treiber, O. D.
Factors in Automotive Diesel Development

White, L. T.
Why Waste Fuel Through the Exhaust?

Apparatus Developed for Measuring Exhaust Pressure-Waves

By E. G. Gunn
Walker Manufacturing Co.

EARLY types of mufflers were usually of the baffle type; that is, the gas went through holes into an expansion chamber, then through holes into another chamber. It is a well-known fact that restrictions offer much more resistance to the flow of exhaust gas than to the passage of sound. Consequently, to get a fair degree of quietness, the restriction type must either have considerable back pressure or be unduly large.

Within the past three years there has probably been more muffler research than for many preceding years. This has largely been due to the demand for quieter cars and to the increase in average traveling speeds.

It would seem that muffler design is an acoustical problem and should be treated as such, but consultation with some of the leading acousticians of the country shed little light on the problem.

Without accurate and simple means of determining what goes on in an exhaust system, the problem is complicated, so an optical magnification manometer was developed for making polar diagrams of pressure waves in an exhaust system. Doubt was expressed by some acousticians as to the possibility of making closed diagrams which could be viewed, because it was thought that the waves would not repeat accurately. This was found not to be the case, as the waves do repeat exactly, as long as the engine runs steadily.

The manometer (Fig. 1) consists of a small stressed diaphragm which oscillates a tiny mirror. When small pressure variations, such as coming from the end of the tail pipe are to be observed, a very thin diaphragm is used. When larger pressure variations are to be observed, a thicker diaphragm is used. Fig. 2 shows the optical arrangement. The image of a pinhole in front of a powerful light is reflected and focused on a screen where it can be observed, traced on tracing paper, or recorded on light-sensitive material. The mirror is tilted slightly, so that the spot of light describes a circle (base circle) on the screen. Variations from this circle are recorded as pressure variations on the diaphragm. A loosely-mounted flywheel serves as a Lanchester dampener to keep the apparatus revolving smoothly.

The manometer is rotated at half engine speed and is connected by tubing to various points in the exhaust system. While the engine is running, the effect of changes in the system or adjustments can be noted on the screen much more accurately than by ear. The instrument is used principally to observe pressure variations of not over 1000 per sec. For high pitch noises a Stewart filter (Fig. 3) is used with a stethoscope. This filter is designed to let through only sounds above 625 vibrations per sec.

The original work with the manometer was done with a small single-cylinder engine as the curve is longer and easier to read. Fig. 4 shows a pressure-wave diagram from the single-cylinder engine. The manometer pipe was connected to the exhaust pipe near the engine and the exhaust pipe was open, i. e., there was no muffler. The number of waves per second agrees with the calculation for an organ pipe of the same length. Although the average pressure was too low to register on a water column, the highest peaks in this diagram represent a pressure of 30 in. of water. Fig. 5 was made by firing a pistol in the end of the pipe, disconnected from the engine, and rotating the manometer by the engine. These two curves show very clearly that pipe resonance can be, and is, excited by a single explosion and continues to sound long after the exciting force has ceased.

Apparently the first note that we have to contend with

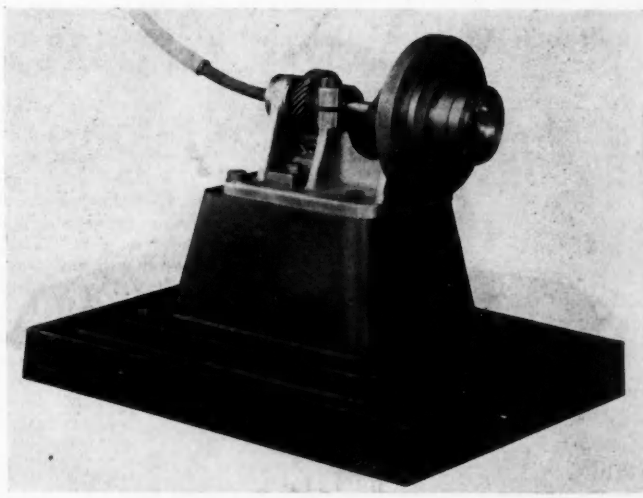


Fig. 1—Manometer arrangement. A small stressed diaphragm oscillates a tiny mirror. When small pressure variations are to be observed the diaphragm is proportionately thin.

[This paper was presented at the Annual Meeting of the Society in Detroit, January, 1934.]

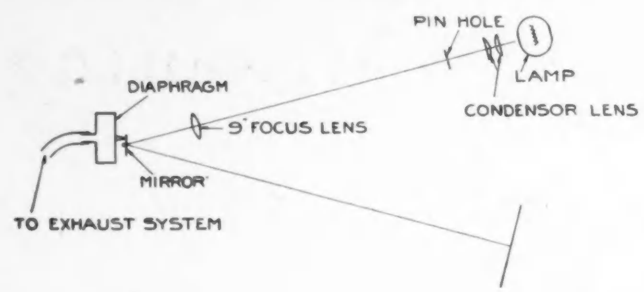


Fig. 2—Optical arrangement. The image of a pinhole in front of a powerful light is reflected and focused on a screen where it can be observed or recorded.

in an exhaust system is the fundamental (and overtones) of the exhaust pipe itself. In many cases observed, the lower critical period of exhaust notes on cars always occurs when the number of explosions per second agrees with the fundamental note of the exhaust pipe. The fundamental of most exhaust pipes is in the order of 60 per sec., while the note coming from the tail pipe is usually much higher and is composed of many frequencies.

Figs. 6-6A to 11-11A show pressure curves at the end of the tail pipe for a $3\frac{1}{4}$ by $4\frac{3}{8}$ in. six-cylinder L head engine. The muffler used for the lower curves was a production through-type muffler which is not satisfactory for this engine, while the one used for the upper curves was a production, wave-interference type muffler, and is satisfactory for this car. The critical speed (about 1200) is clearly shown. The number of explosions at this speed is the same as the fundamental of the exhaust pipe.

The noise observed in a car usually seems worse than that observed on the dynamometer, even though the pipe is run through a brick wall into a comparatively quiet place outdoors. Pressure or sound waves are traveling back and forth in the system, even back against the exhaust stream to the manifold. It is possible that the chassis under these conditions is vibrated by the exhaust system and acts much the same as a sounding board on a piano.

Exhaust noise is not necessarily carried along the gas stream. In Fig. 12 is shown a system with a removable muffler shell. Fig. 13 shows sound curves taken at the end of the tail pipe of a six-cylinder engine with the outer shell on and off. Although there is a high velocity of gas flowing out of the tail pipe with the outer shell removed, the sound coming out is so slight as to scarcely be heard with a stethoscope.

Noise elimination as applied to mufflers can be classified roughly as being accomplished in two ways:

1. Sound wave absorption
2. Sound wave cancellation (interference)

(1) Mufflers which depend on sound absorption convert sound energy into heat. This can be done by the use of Helmholtz resonators, sound absorbing chambers such as Stewart filters, by friction of the waves passing through small holes in the through tube to the surrounding chamber, or by the use of porous material which absorbs sound by friction of the waves passing through.

(2) Mufflers which depend on wave interference are of three general types:

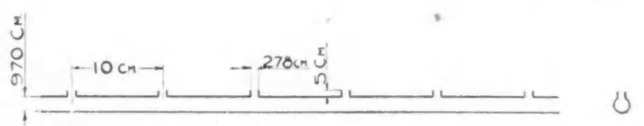


Fig. 3—Stewart filter for high pitch noises. This filter is used with a stethoscope and is designed to let through only sounds above 625 vibrations per sec.

(a) Fig. 14—Ones which incorporate Quincke tuners, i. e., a closed-end tube of a length equal to one quarter the wave length of the wave to be cancelled. The wave enters the tube, is reflected from the far end, and returns out of phase to completely or partially cancel.

(b) Fig. 15—Ones which incorporate Herschell tubes, i. e., a divided path for the sound waves, one path being one-half of the wave length to be cancelled longer than the short path. The waves, on re-uniting one-half wave length out of phase, cancel.

(c) Fig. 16—Ones in which the gas travels through perforated pipes, alternately from one end of the muffler to the other. These are usually made with three pipes, the gas first flowing from the front to the rear through one pipe, then through another pipe to the front, and then to the rear and into the tail pipe. At any transverse section of the muffler a wave in one pipe is out of phase with the wave in another

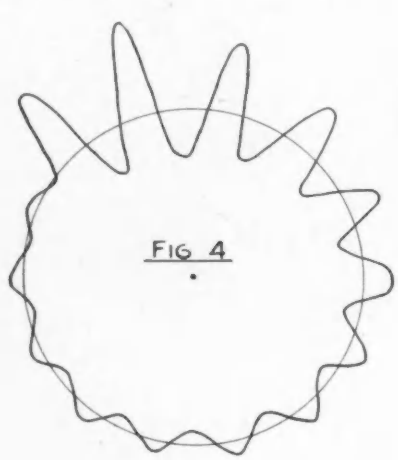


Fig. 4—Pressure-wave diagram from single-cylinder engine. The manometer pipe was connected with the exhaust pipe near the engine. No muffler intervened.

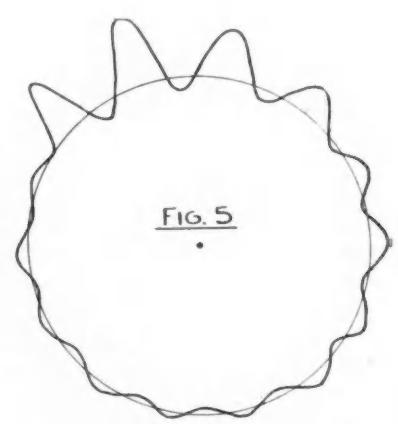
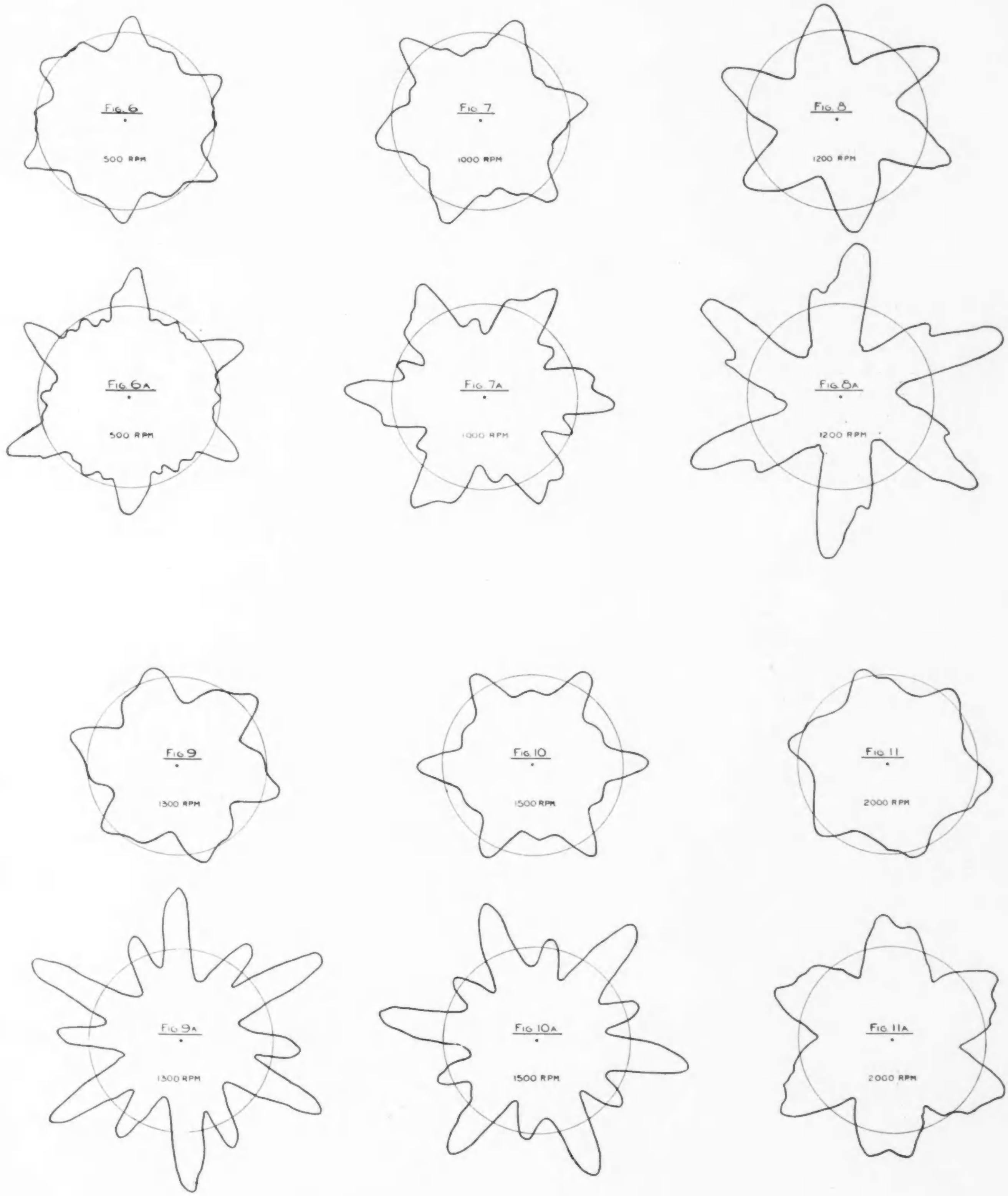


Fig. 5—Diagram made by firing a pistol in the end of the exhaust pipe disconnected from the engine, while the manometer was being rotated by the engine.



Figs. 6-6A to 11-11A—Pressure curves at the end of tail pipe for a 3¼ by 4⅜ in., 6-cylinder, L-head engine. The muffler used to obtain the lower set of curves was a production through-type not satisfactory for the particular engine. The upper curves were obtained from a production wave-interference type muffler which was statisfactory for the engine.

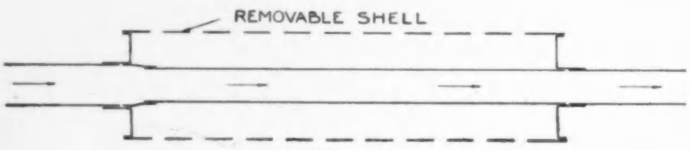


Fig. 12—System with a removable muffler shell.

pipe, can short circuit across the surrounding chamber and cancel.

There seems to be a great difference in tail pipe sound between muffler tubes having various kinds of holes. Fig. 17 shows two muffler tubes, "A" having round holes and "B" having tangential louvers. Fig. 18 shows a sound picture taken at the end of the tail pipe.

Back Pressure

There is a wide difference of opinion as to what is a proper back pressure. Top-speed back pressures have been observed as low as 4 in. of mercury and as high as 15. There are probably some cars above and below these figures. Although lowered back pressures show increased horsepower on the dynamometer, lowering much below 6 in. shows little difference in speed on the road. Where back pressure begins

FIG. 14

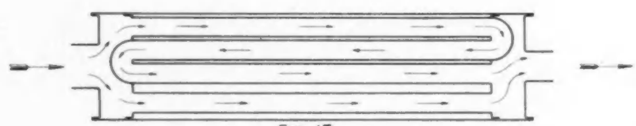


FIG. 15

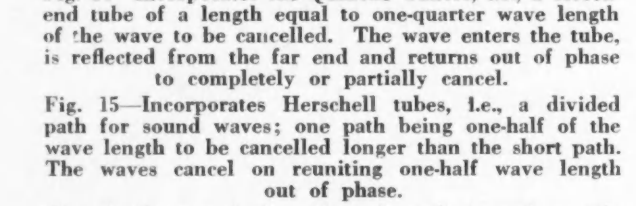


FIG. 16

Figs. 14 to 16—Three general types of wave-interference mufflers:

Fig. 14—Incorporates the Quincke Tuners, i.e., a closed-end tube of a length equal to one-quarter wave length of the wave to be cancelled. The wave enters the tube, is reflected from the far end and returns out of phase to completely or partially cancel.

Fig. 15—Incorporates Herschell tubes, i.e., a divided path for sound waves; one path being one-half of the wave length to be cancelled longer than the short path. The waves cancel on reuniting one-half wave length out of phase.

Fig. 16—Gas travels through perforated pipes alternately from one end of the muffler to the other. At any transverse section of the muffler a wave in one pipe is out of phase with the wave in another pipe and can short circuit across the surrounding chamber and cancel.

to affect valve life, and to what extent, apparently has not been accurately determined as yet. Low back-pressure, and at the same time quietness, means more room, more weight, and more cost, and these go up rapidly as top-speed back pressures go much below 6 in.

It might be observed here that mercury columns do not tell the whole story in regard to back pressures. Many cases have been observed in which a change involving an increase in back pressure has resulted in an increase in power. This is due to the fact that the wave form is such that the pressure is lowered at upper dead center. This is a field in which little work has been done.

Mountings

Until recently, mufflers were mounted rigidly on the frame, but as the other parts of the car were made quieter, noises

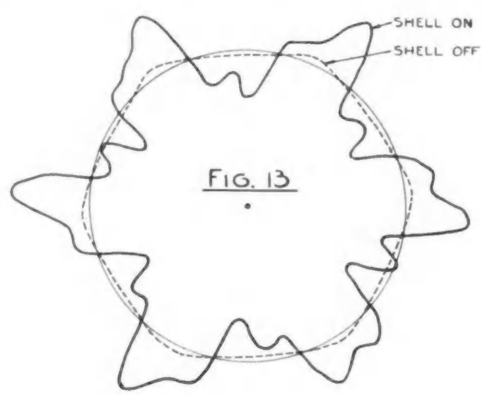


Fig. 13—Sound curves taken at end of tail pipe of a 6-cylinder engine with and without muffler shell.

due to muffler mounting began to show up. It is sometimes difficult to tell whether an observed noise is due to muffler mounting or something else. To segregate mounting noise from other noises, a trailer (Fig. 19) was built. An observed noise which disappears when this is used, is due to muffler mounting, and can be practically eliminated by changes in muffler mounting. One successful mounting consists of attaching the exhaust pipe to the rear end of the powerplant, letting the front end of the muffler hang on the exhaust pipe and supporting the rear end of the muffler on a bracket carried in soft rubber.

Shell Noise

So-called "shell noise" probably originates at the exhaust valve and is caused by the action of the exhaust gas traveling at high velocity through the partly opened valve. This high pitch noise causes a vibration of the whole exhaust system and can be heard with a stethoscope as coming through the walls of exhaust pipe, muffler shell and heads, and some through the walls of the tail pipe.

Overhead-valve engines are considered the worst offenders, but sometimes a muffler which has no objectionable shell noise on one overhead-valve engine, will have a pronounced shell noise when used with another overhead-valve engine.

Obviously, the ideal condition would be to prevent this noise by shaping the valves and seats so that the released pressure traveling at high velocity through the small open-

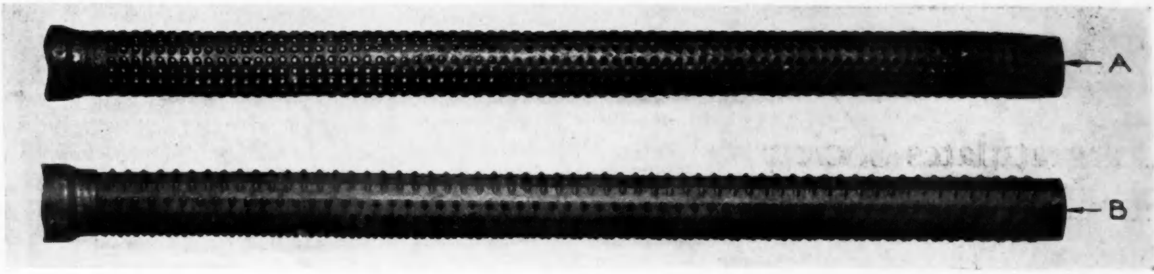


Fig. 17 — Muffler tubes; one has round holes (A) and the other tangential louvers (B). The author concludes that there is a difference in tail-pipe sound with mufflers having various kinds of holes.

ing would not cause a vibration. Experiments carried out along this line have so far done little. The next best thing then would be to silence this noise before it gets into the exhaust pipe.

There has appeared lately a muffler in which the passage of part of the exhaust gas through a venturi creates a lower-than-atmospheric pressure to be used for operating the windshield cleaner, horn, fuel system and possibly other things. It is easy to get upwards of 12 in. of mercury of vacuum. Fig. 20 shows one design. In this arrangement, the back pressure at the lower speeds increased over that of a through type muffler, but not necessarily at high speeds. The higher back-pressure at the lower speeds very materially helps silencing. As the speed increases the resistance and velocity-head pressure on the valve moves it against the spring pressure to the right, by-passing part of the exhaust gas. By-passing part of the gas is done in much the same manner that intake gas is by-passed in an air-valve carburetor.

I am indebted to the Nash Motors Co. for the use of one of their engines and dynamometer.

(Discussion of Gunn paper begins on following page.)

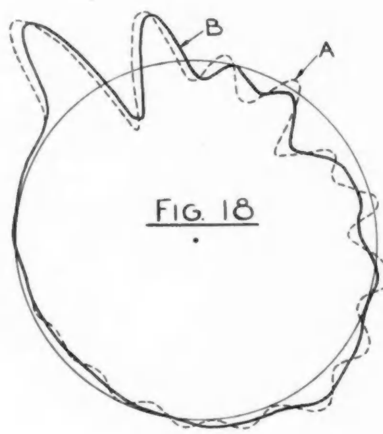


Fig. 18—Sound picture taken at the end of pipe with the tubes shown in Fig. 17.

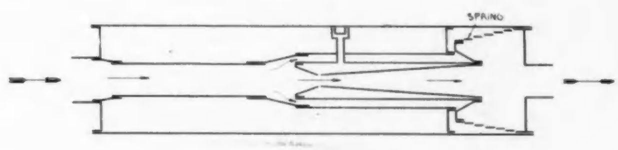


Fig. 20—Muffler design in which passage of part of the exhaust through a venturi creates a lower-than-atmospheric pressure to be used for operating windshield wiper, etc.

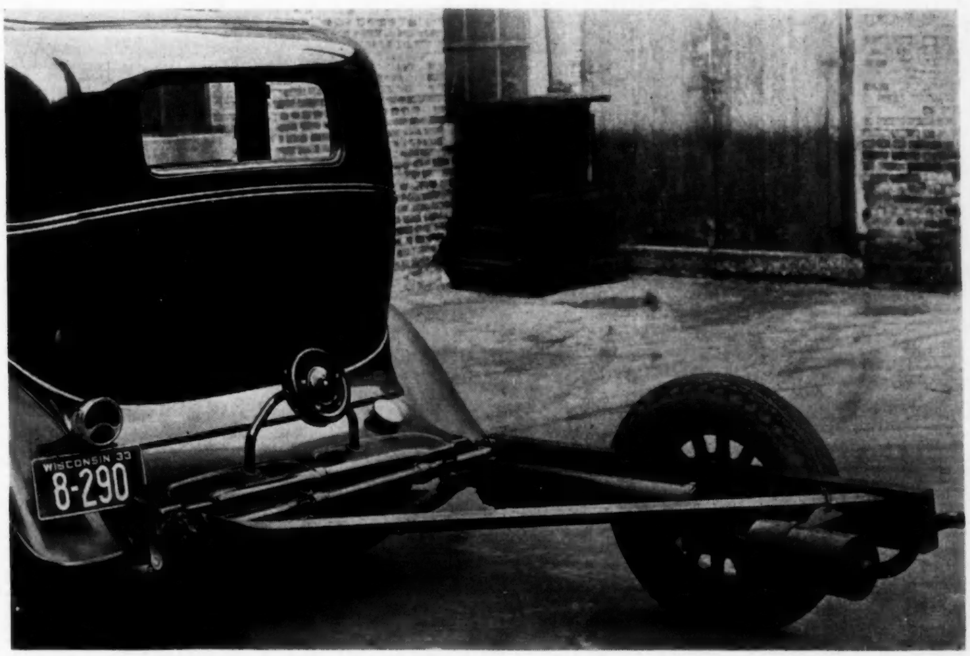


Fig. 19—Trailer arrangement built to segregate mounting noise from other types of noise.

Discussion

Congratulates Society For Tackling Problem

—B. J. Lemon
U. S. Tire Co.

THE automobile can be considered as composed of a number of acoustic resonators, having varied degrees of coupling between them.

Automobile noise, though useful as a detector of mechanical operating imperfections, is otherwise extremely undesirable, and its sound disagreeably out of place.

This sentiment has been developed steadily in the minds of automotive engineers and is fast taking shape in the minds of the public.

The noise problem in the automobile arises from the fact that noise is objectionable, that it produces certain disagreeable effects upon people such as annoyance, interference with hearing, effects upon driving efficiency, and upon health.

Some automobile noises are more annoying than others; the degree seeming to depend upon whether the noise is quite unnecessary, such as squealing tires or brakes, body rattles or exhaust or intake pulsations.

It is reasonable to assume that a quarter of our population is subjected to automobile noise, daily, during their working hours.

The noise problem is a big one, wherever tackled, and this includes the automobile. May today's session on noise be the forerunner of others to come, so that those of us who are compelled to virtually live in our cars may thank this Society for courageously taking another advanced step in the direction of greater riding-comfort for the motorist.

Says Two Misconceptions Implied in Presentations

—E. J. Abbott
Research Physicist, Engineering Research Department, University of Michigan

IN the course of this symposium two misconceptions have been either stated or implied several times. These two misconceptions are:

1. That the loudness of sounds as heard by ear cannot be determined from instrumental measurements.

2. That soundmeters are a needless refinement in noise reduction work because the ear is the final judge so that the ear is a sufficient tool for attacking noise-reduction problems.

Both of these statements are wrong. If sound measurements are taken so that the physical quantities, i. e., the sound pressures and the frequencies, are known, these measurements can be interpreted in terms of known data on human hearing so that the loudness as it appears to the ear can be determined far more accurately than it can be judged by an individual observer. For convenience, sound pressures are usually expressed in terms of decibels above some reference sound pressure, but this does not change the fact that the measurements are definite physical pressures.

1. In the case of individual notes, data on the variations of the sensitivity of the average human ear with frequency and with level have been available for nearly seven years.[1] Recently, these data have been completely checked and extended.[2] The data which have been in use for seven years check almost perfectly with the latest data when they are both referred to the newly determined threshold curve which forms part of the latest data.

In the case of complex sounds, a formula has recently been published for computing the loudness of sounds as heard by ear from physical data obtained by meter measurements.[3] A more rapid method, which appears to have great practical value, is the "total noise" meter, in which the various component frequencies of a sound are weighted according to the sensitivity of the ear for sounds of the level being measured. Checks of this type of meter against the judgments of groups of observers on a wide variety of sounds show a truly remarkable agreement.[4, 5] In this connection, it is important to note that the average judgments of two groups of observers agree with themselves and with the meter with surprising accuracy, but that individual observers disagree widely with this average. Consequently, the loudness, as it would be heard by a group of observers, can be determined far more accurately by meter than it can be by ear listening of any single observer. The essential items in loudness determination are accurate physical measurements, properly interpreted in terms of data on ears. Failure in either of these points results in absurdities.

2. It is a fact that single notes are seldom encountered in practice, so that practical noise reduction problems deal with complex sounds. It is also a characteristic of the human sense of hearing that if a sound consists of several components of approximately equal loudness, the removal of any one of these components produces a very minor change in *loudness*. This can easily be demonstrated either by meter or by ear, and the amount of the change to the ear can be determined by meter readings. It is also true that a removal of one component of a sound usually produces a noticeable change in the *quality* of the sound, which may or may not be important, but which does add to the difficulty of judging loudness by ear.

In noise reduction work, it is necessary to bring about reduction in *all* the components of the sound which contribute essentially to the loudness before any practical reduction is obtained. The best way to do this is obviously to determine the relative importance of the various components, and since, as explained above, this cannot be done by ear, the only logical method of noise reduction is by the use of soundmeters. In this way, the importance of the various components of the sound can be determined, and the reductions brought about by any change can be measured. In this way, the loudness of the entire sound can be reduced by a combination of changes, any one of which would be negligible in itself but is essential to the whole. After the overall reduction is obtained, it can easily be distinguished by ear, or

[1] Direct comparison of the Loudness of Pure Tones by B. A. Kingsbury, *Physical Review*, April, 1927, p. 598.

[2] Proposed Standards for Noise Measurement; A. S. A. Report, June, 1933; *Journal Acoustical Society*, October, 1933, p. 109.

[3] Loudness, Its Definition, Measurement, and Calculation by Harvey Fletcher and W. A. Munson, *Journal Acoustical Society*, October, 1933, p. 82; *Bell System Technical Journal*, October, 1933, p. 377.

[4] On the Loudness of Noise by H. B. Marvin, *Journal Acoustical Society*, January, 1932, p. 388.

[5] Sound Measurements vs. Observers' Judgment of Loudness by P. H. Geiger and E. J. Abbott, *Electrical Engineering*, December, 1933, p. 809.

measured by meter, but the ear is practically useless for carrying out the steps to this end. These steps can be made logically and definitely with the aid of meter measurements.

Interprets Some of Data In Form of Different Law

—F. C. Mock
Eclipse Aviation Corp.

I WANT to applaud the author's work and his ingenious method of attack, and particularly the progress he and others are making in destroying the old notion that exhaust silencing can only be obtained with back pressure. It seems important, however, that attention should be called to what I am sure is an error in the basic concept here adopted. Nearly every one will agree that the individual "cracks," which we hear from each cylinder of an engine running without an exhaust manifold, are not reverberating sounds. They are not like the vibration of a violin string, or ripples on water; they are rather individual tidal waves, initiated by gas rushing out at the velocity of sound in atmosphere or higher, like the rush of gas from a dynamite explosion.

The recurrent waves of large magnitude shown on the diagrams of the paper only occur when a pipe of some length is applied to the exhaust port. The fallacy of using their interference to deaden the sound is shown by considering that any second wave cannot start its phase, rush back into the pipe and out again, until after the primary impulse has reached the atmosphere and is on its way to the listener's ear. I am sure that the success which the author is having is due to a different action, of which the law may be expressed as follows: "With a single wave, after it has left its pressure source, the energy potential across the wave front is inversely proportional to the area or spread of the wave front."

In the cases illustrated, the initial wave front energy is dissipated into the side orifices and chambers, and not returned until the main impulse thus diminished has reached the outside air.

There are of course pipe waves, audible as "roar," "throb" or "beat"; also as described, the initial "cracks" may transmit to the outside air through the exhaust manifold or the exhaust pipe walls. Also rough edges near the outlet of the system may generate noise from an otherwise silent gas flow; for example, hold a coarse file at the outlet of a muffler tail pipe.

Conversion to Velocity Increase

The paper states that the exhaust sound may be reduced by back pressure or by wave interference. We have also converted it into velocity increase by the same principle as that used in the efficient expansion of steam through turbine nozzles; so that the gas comes out of a 2-in. muffler outlet silently, and both cooler and more forcibly, than from the 2-in. exhaust pipe with the muffler removed.

This particular conception of the manner in which the exhaust wave is generated by the cylinder pressure and exhaust valve opening, with some diagrams similar to those shown by Mr. Gunn, is given in a paper by myself printed in the 1914 S.A.E. TRANSACTIONS, Part 1, page 69.

New Color Methods Aid to Car Sales

Since the use of color as a merchandising factor in the sale of automobiles was introduced, its importance has swept other industries. With the help of color in advertising, in the movies, in architecture and in clothes, we have become a color conscious people. Were it not for the sobering influence of the economic situation, it is highly probable that car color preference would embrace many exotic variations at this time. Even so, the appeal of beauty is most highly regarded as a sales stimulus today. Beauty is comprised of harmony in form and color.

Five or six years ago, many cars were finished with the upper structure in black or dark colors contrasting with lower areas in light color values. You will recall that cars were very much higher than they are now, and this mode of color distribution was intended to engender an aspect of lowness. The dark color was regarded as the "heavy" color and was thought to give an impression of weight, visually lowering the car silhouette.

More recently the error of this line of reasoning has been disclosed. A study of the color masterpieces of the world in ceramics, rugs, paintings, and tapestries, as well as color distribution in nature, reveals a consistent color relationship of high, middle and low values with high value colors at the top and low value colors at the base. The high or light value color is the sky, the middle value color is the horizon and the low or dark value color is the foreground in nature's landscape. This relationship is also carried out in rug color arrangements, vases and the like. The dark color represents weight and is used to direct the eye toward the edges or bottom of the design. On an automobile, lower body areas finished in darker colors than are used on the upper portion create an aspect of greater safety by strengthening the impression of a low center of gravity and eliminating the top-heavy look, which invariably results from placing a large mass of a dark color on top of a light color. That is why we have produced so many cars finished in this manner in late years.

Color selection used to be a matter of time-consuming perplexity. The many thousands of colors offered by the 250 concerns making motor car lacquers lacked rhythm and system in their presentation. Those of you who have escaped the ordeal can imagine the labor and pains required to sort over stacks of unrelated color variations in search of suitably harmonious color combinations. Put upon this the necessity of selecting upholstery fabrics and trim colors which will blend and not clash with the body colors and the problem is further complicated.

Since the S.A.E. has undertaken color classification and control, as indicated by the announcement of the Standards Committee appearing in the S.A.E. JOURNAL for December, we can hope for greater satisfaction and economy in color presentation of the future. The system of color gradation Duco Colors offered as a practical solution to production color requirements is proving of real merit to engineers.

—Digest of paper written for 1934 Annual Meeting by W. G. Haarz, Jr., Graham-Paige Motors Corp.

Rotating-Wing Aircraft Compared to Conventional Airplanes

By John B. Wheatley

Junior Aeronautical Engineer,
National Advisory Committee for Aeronautics

THIS paper contains a brief discussion of the advantages inherent in a rotating-wing aircraft in regard to its performance and safety as compared to the conventional airplane. An arbitrary criterion is set up that presents the different characteristics which must be possessed by the rotating-wing aircraft in order that it be considered successful and practical, and the criterion is then used to evaluate the merit of the helicopter, the cyclogiro, the autogiro, and the gyroplane.

According to the criterion, the autogiro and gyroplane will be superior to the airplane when their pronounced possibilities for high-speed performance are materialized, these possibilities consisting of the inherent ability of the autogiro and gyroplane rotors to attain their maximum lift-drag ratio at any desired forward speed. The cyclogiro is approximately equal in merit to the airplane, while the helicopter is quite definitely inferior.

The discussion indicates the utility of the rotating-wing machine for the private flyer and the unskilled pilot, because of its increased safety and the smaller landing field required for it.

ALMOST all the hazards encountered in flying an airplane are connected with the phenomenon of a gradual weakening of control as the flying speed approaches its minimum. As minimum speeds range from 50 to 75 m.p.h. an undesirable premium is placed upon piloting technic during landings and take-offs. A rotating-wing aircraft suffers very slightly from these handicaps because the relative velocity of the lifting surfaces to the air is independent of the translatory velocity of the machine and is always large, so that the angle of attack of the lifting surfaces is well below the burble point. The resultant performance of rotating-wing aircraft thus materially extends downward the low-speed phase of flight, lessening the piloting skill required for emergency landings and take-offs, and making the pilot more independent of meteorological conditions because at low speed a shorter visibility is required for the same degree of safety.

Many difficulties lie in the path of the engineer who attempts to realize the advantages inherent in the rotating wing, a statement which is attested by the hundreds of unsuccessful designs that have appeared in the aeronautical world. There have been innumerable attempts to build a heavier-than-air machine that would rise and descend vertically and remain stationary in the air, and a smaller number of attempts to obtain less ideal performance by employing a rotating wing for lift.

The marked advantages in slow-speed flight, control, and safety possessed by the rotating wing have attracted the attention of several of the well-known aeronautical research organizations, as well as the interest of many individual inventors. The National Advisory Committee for Aeronautics has for several years been conducting a program of research to examine and improve existing types of rotating-wing aircraft and has been encouraging the development of any new type that showed promise. Among its achievements in this program are included the procurement of the first detailed experimental information on the aerodynamic characteristics of the autogiro rotor[1]; and the aerodynamic analysis of the gyroplane and cyclogiro rotors[2].

[This paper was presented at the Annual Meeting of the Society in Detroit, January, 1934.]

[1] Lift and Drag Characteristics and Gliding Performance of an Autogiro as Determined in Flight; see N.A.C.A. 1932 Technical Report No. 434; and Wing Pressure Distribution and Rotor Blade Motion of an Autogiro as Determined in Flight; see N.A.C.A. 1933 Technical Report No. 475.

[2] An Aerodynamic Analysis of the Gyroplane Rotating-Wing System; see N.A.C.A. Technical note; and Simplified Aerodynamic Analysis of the Cyclogiro Rotating Wing System; see N.A.C.A. 1933 Technical Note No. 467.

A series of N.A.C.A. wind-tunnel tests is now being made in which models of the gyroplane, autogiro, and cyclogiro will be tested. Although the discussion in this paper will be confined to the rotating-wing system of the helicopter, cyclogiro, autogiro, and gyroplane, the N.A.C.A. has examined and encouraged other promising types, one of the most unusual, the vertaplane, being a biplane whose upper wing can be released and allowed to rotate in the manner of an autogiro rotor.

Criterion for Comparison

In order to compare the relative merits of the various types of rotating-wing systems, an arbitrary set of conditions will be set up which a successful rotating-wing machine must satisfy. In the application of this criterion, the standard of comparison will be the airplane, and the merit of the rotating wing will be established according to the degree by which it exceeds or falls short of the characteristics of the airplane.

The criterion consists of a set of six conditions, as follows:

1. Controllability, maneuverability, and stability must be positive and satisfactory over the entire speed range.
2. The machine must be reliable, and must be able to land safely in the event of a powerplant failure.
3. The low-speed performance of the machine must permit landing on and taking-off from small, restricted areas.
4. The high speed of the machine must be great enough for practical utility.
5. The operation of the machine must be simple, so that a comparatively inexperienced pilot will be able to operate it.
6. First cost and maintenance must be small, and the operation must be economical.

The first and second conditions stated are extremely important; in fact, any serious deficiency in either one is sufficient to make the machine under consideration entirely impractical.

The Helicopter

The first type of rotating-wing aircraft which will be taken up is the helicopter: a flying machine that normally derives the major portion of its sustentation from one or more power-driven lifting propellers rotating about an approximately vertical axis. It is safe to say that more diverse attempts have been made to build a machine of this type than any other form of rotating-wing machine, and only a very small number of such attempts have been even moderately successful. Figs. 1 and 2 show two photographs of helicopters which were successful to the extent of leaving the ground. More complete discussion of these types and other helicopters than may be given here can be found in published material[3]. As an illustration of the complexity of the type, the Oehmichen helicopter employed twelve propellers; four for sustentation, two for propulsion, and six for control and stability. In view of the almost equal complexity of the other designs which have been built, it can be stated that the helicopter is inherently a complicated mechanism. Further examination shows that all other phases of flight have been sacrificed to the low-speed or stationary-flight conditions, which materially reduces the utility of this type. The criterion applied to the most satisfactory helicopter gives:

1. The complexity of the control system required and the change in type of control during different stages of flight will result in a slight deficiency in control and maneuverability.
2. The complication inherent in the helicopter reacts against its reliability, and emergency landings may be difficult.
3. The low-speed performance is much better than that of the airplane.
4. The high speed is poorer than that of the airplane because of the excessive parasite drag and inefficient lifting surfaces.
5. The operation of the machine will be complex and require skilled technic.
6. First cost, maintenance, and operating cost will be high.

The Cyclogiro

The cyclogiro is of recent origin; too recent, in fact, to have been demonstrated at full scale. A diagrammatic sketch of the machine is shown in Fig. 3. Briefly, the cyclogiro consists of a fuselage of conventional form, supported by two power-driven paddle-wheel rotors, one on each side, rotating about the lateral axis. The cyclogiro rotor consists of three or more blades which are oscillated by cams during rotation in such a manner as to develop a resultant force; the direction and magnitude of the resultant force are controlled by adjusting the cams so as to alter the amplitude and the phase angle of the oscillation with respect to the direction of flight. Independent control of the two rotors enables the pilot to generate a yawing or rolling moment, and the center of gravity of the fuselage is in such a position that the moment of the fuselage weight counteracts the rotor torque.

[3] An Introduction to the Helicopter by Alexander Klemin; see N.A.C.A. 1927 Technical Memorandum No. 340.

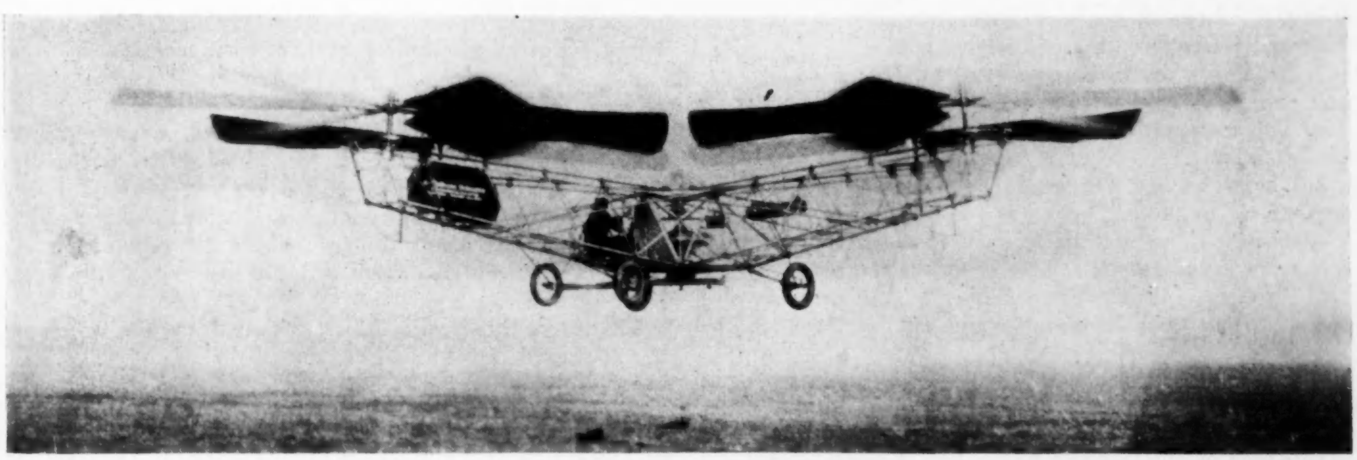


Fig. 1—The DeBothezat Helicopter

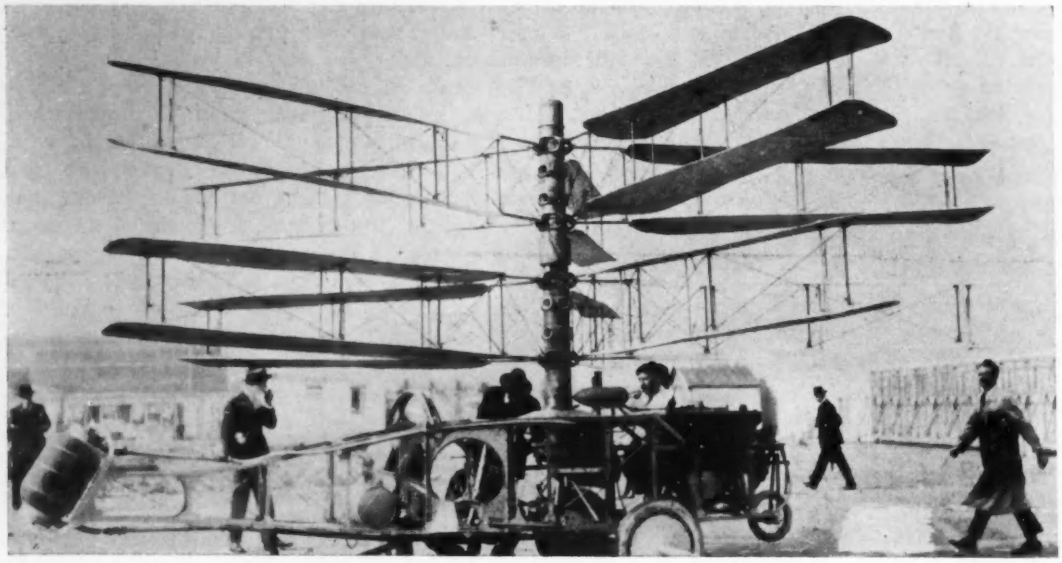


Fig. 2—The Pescara Helicopter

In the event of a powerplant failure, a suitable choice of cam position will enable the rotor to obtain driving torque during a part of the revolution from the air forces acting upon it, and during the remainder of the revolution sustaining forces will be developed by it. An analytical examination of the machine demonstrated this statement satisfactorily, and wind-tunnel tests now being conducted by the N.A.C.A. on an 8-ft. diameter model of the rotor will supply experimental evidence concerning the relationships expressed in that analysis.

The present lack of experimental data on this machine reflects some doubt upon any quantitative statements made concerning its merit; however, a criticism in general terms can be considered valid. As seen from the sketch, there is an appreciable amount of structure that adds to the parasite drag of the machine. This arrangement necessarily handicaps the performance in comparison with the conventional airplane. However, there is a gain in propulsive efficiency arising from the fact that both propulsive and sustaining forces are generated by the rotors, which eliminates the power losses occurring in the propeller of the conventional airplane. It is considered that this gain will practically offset the losses caused by the inherently higher parasite drag.

Evaluating the criterion, the following values appear possible of attainment:

1. Control and maneuverability at low speed will be excellent, and better than that of the airplane.

2. Reliability should be equivalent to that of an airplane, and emergency landings will be easier.

3. The low-speed performance is much better than that of the airplane.

4. The high speed will be of the same order as that of the airplane.

5. The machine will be slightly more difficult to pilot because of the additional complexity in the control system.

6. First cost will be slightly higher but maintenance and operating cost will be about equivalent to those of the airplane.

It will be noted that the only unfavorable comment in the criterion was concerned with the complication inherent in the machine, which is caused by the presence of an additional control for the rotors. This control would probably be manifested as an auxiliary control stick attached to the main one and would be used in the variation of the magnitude and direction of the resultant force generated by the rotors. The penalty is offset, however, by the greatly improved low-speed performance, which makes it possible for the machine to remain stationary in the air and to land after a vertical descent in a very small and restricted area.

The Autogiro

The autogiro is perhaps the best known of all rotating-wing aircraft because of fairly extensive use and intensive advertising. It is the result of the genius of Juan de la Cierva, who invented the principle upon which it works and has carried forward its development to a remarkable position in the last few years. The American version of the autogiro has been produced by Harold F. Pitcairn, who is responsible jointly with Cierva for much of the recent technical advancement of the machine.

The autogiro may be defined as a heavier-than-air machine that generates the principal part of its lift by a rotor having three or four blades which rotate freely under the action of air forces about an approximately vertical axis. In forward flight the thrust of the rotor on opposite sides of the plane of symmetry is equalized by a free flapping motion of the blades about hinges near the axis of rotation, which permit the blades to oscillate in planes containing their span axes and the axis of rotation. The flapping hinges also prevent the transmission of precessional moments from the rotor to the fuselage and simplify the structure of the blades by resolving their major stresses into tension. In the latest models, control in pitch and roll is obtained by tilting the rotor axis about pivots through which the resultant force of the rotor passes. A picture of one of these recent machines is shown in Fig. 4.

The pronounced possibilities possessed by the autogiro have attracted the attention of most of the large aeronautical research organizations. The results of considerable work have been published in Germany by the D.V.L., in England by the British A.C.A., and in the United States by the N.A.C.A. It may be stated that the only thorough full-scale work has been accomplished by the N.A.C.A.,[4] and the

[4] Lift and Drag Characteristics and Gliding Performance of an Autogiro as Determined in Flight; see N.A.C.A. 1932 Technical Report No. 434; and Wing Pressure Distribution and Rotor Blade Motion of an Autogiro as Determined in Flight; see N.A.C.A. 1933 Technical Report No. 475.

results of this work have been of great assistance to both the aerodynamic and the structural engineer. The fact of primary importance which has been brought out by previous investigations is that present-day performance of the autogiro is far inferior both to what may be obtained from the conventional airplane and to what may be expected from the autogiro after some additional development work.

The most important deficiency of the present autogiro is its high-speed performance. For this reason a detailed discussion of autogiro high speed will now be given. The essential cause of the poor maximum speed of the autogiro is its excessive parasite drag, an item which is to a great extent controllable by the designer. Defining the parasite-drag area of an aircraft as an area which, when multiplied by the dynamic pressure, results in the parasite drag, the criterion of the weight divided by the parasite-drag area can be used as a measure of the aircraft's cleanness. Several modern airplanes have attained a value of this criterion of 1000 lb. per sq. ft. merely by retracting the landing gear and by careful fairing of the wing roots and engine nacelles. The best value attained on an autogiro without the above refinements has been about 300 lb. per sq. ft.; however, good design, attention to detail, and some alteration of the lines would raise the autogiro figure to 900 lb. per sq. ft. without any radical departures from present practice. As an example, consider a four-place autogiro of 3000 lb. gross weight. The complete structure would consist of the rotor, cantilever pylon, a fuselage and engine, a retractable landing gear, and fixed tail surfaces. Experience indicates that the drag areas of the component parts would be:

Item	Drag Area
Fuselage and engine	2.5 sq. ft.
Pylon	0.4 sq. ft.
Tail surfaces	0.2 sq. ft.
Total	3.1 sq. ft.

$$\frac{\text{Weight}}{\text{Drag Area}} = \frac{3000}{3.1} = 968 \text{ lb./sq. ft.}$$

The autogiro rotor has at the present time been developed only to the point that indicates its ultimate possibilities. The majority of rotors in use in this country have a maximum lift-drag ratio of from 8 to 10 which occurs at a speed of about 100 m.p.h., though it is easy to show that values of from 12 to 15 may be obtained at higher speeds without serious difficulty, and without materially sacrificing the low-speed performance. The analytical expression for the lift-drag ratio of a rotor is derived by considering the energy losses in a rotor, which arise from two, and only two, sources: the profile drag of the blade elements; and the induced losses associated with the generation of lift. The theoretical derivation of the exact expression for the lift-drag ratio is possible[5], but as a working conception of the problem can be obtained from a general discussion, the long and involved derivation will not be presented.

It is evident that the rotor-induced drag at high speed is small in comparison to its profile-drag losses, since the induced drag is inversely proportional to the square of the speed, as for a wing. Calculations have indicated that the fraction of the total drag-lift ratio contributed by the induced drag is of the order of 5 per cent or less of the total. Analyses have shown that, by employing a more efficient airfoil for the blade profile, the drag of the rotor can be reduced without reducing the lift, and the lift-drag ratio increased as much as 30 per cent. The lift-drag ratio would then be of the order of 13.

It is of considerable importance that this value of the lift-drag ratio can be obtained at a small lift coefficient, and consequently at a large forward speed, by decreasing the ratio between blade area and disc area (the solidity) without changing the blade pitch, disc loading, or rotor diameter. A decrease in the single parameter of blade area then means that the angles of attack of the blade elements corresponding to an equilibrium condition of steady autorotation are unaltered, though the unit loading on each blade has been increased. With a higher unit loading and the same angle of attack the blade rotates faster, and since the maximum lift-drag ratio occurs at a constant value of the ratio between forward speed and tip speed, the forward speed corresponding to maximum lift-drag ratio has been increased in proportion to the tip speed. This property of the autogiro rotor is an extremely valuable one when a large speed range is desired, because the maximum lift coefficient of the rotor changes much more slowly than the solidity. It is then possible to have a maximum lift coefficient of 0.900 and still obtain a lift-drag ratio of 12 at a lift coefficient of 0.0171, corresponding to speeds of 28 m.p.h. and 200 m.p.h., respectively, for a normal loading of 1.75 lb. per sq. ft. on the rotor. A conventional wing having a loading of 9.6 lb. per sq. ft. and a maximum lift coefficient of 1.500 has a minimum speed of 50 m.p.h. and a lift-drag ratio at 200 m.p.h. no larger than 9. The difference between the rotor and wing lift-drag ratios at 200 m.p.h. represents a difference of 44 hp. required for a 3000-lb. machine in favor of the autogiro. The properties of the autogiro are definitely such as to make it better adapted than a wing to the attainment of a large speed range, and from the above analysis, better suited to obtaining a high speed when the high speed is in excess of four times the minimum speed.

There is apparently ample reason for anticipating the development of an autogiro in the near future that will equal or exceed the high-speed performance of the equivalent airplane, that is, an airplane of the same power and useful load. On this basis the criterion for the autogiro will be evaluated by considering, without too much optimism, this future autogiro:

1. The low-speed control is superior to that of the airplane.
2. The reliability is equivalent to that of an airplane, and emergency landings will be easier.
3. The low-speed performance is superior to that of an airplane.

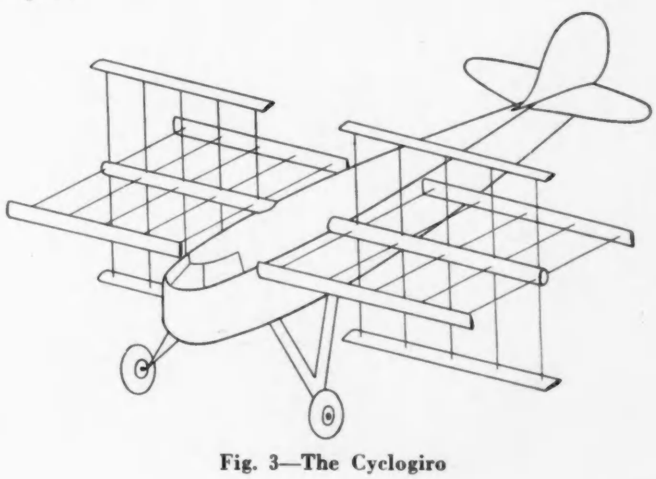


Fig. 3—The Cyclogiro

[5] Further Developments of Autogiro Theory, Parts I and II by C. N. H. Lock, British A.R.C.; see 1927 Reports and Memoranda No. 1127.

4. Airplane high speeds will probably be exceeded.

5. Control system is as simple and easy to use as that of the airplane.

6. First cost will be slightly higher, but maintenance and operating costs will be equivalent to that of the airplane. The criterion shows that in the opinion of the author the autogiro is superior to the airplane in all respects except the last two, and even in the last two there is no serious deficiency. Such a statement will probably be criticized because of the poor high-speed performance of present autogiros, but recent demonstrations of Cierva's new machine have satisfactorily demonstrated that airplane high speeds are definitely not impossible of attainment by the autogiro.

The Gyroplane

The gyroplane has been sponsored by E. Burke Wilford, who has been responsible for several years of research and development on this type. Unfortunately, the potentialities of this machine have not been so clearly demonstrated as have those of the autogiro, though a thorough analysis of the principle upon which it works shows that its merit is high.

The gyroplane is an aircraft that develops the major part of its lift by a freely rotating four-bladed rotor, in which opposite blades are rigidly interconnected and attached to the hub in bearings that permit the blade pair to rotate freely about an axis approximately parallel to the span axes of the blades. The axis of rotation of the rotor is approximately vertical. In forward flight the lift on opposite sides of the plane of symmetry is equalized by a rotation of the blade pair in the hub bearing, which is governed by a controllable cam in the hub mechanism. Pitching and rolling control are obtained by altering the rotation or feathering of the blade pair in the hub bearings with the same cam, the control moment being applied about an axis inclined to the axis about which the motion is desired, so that the precessional moment of the rotor is utilized. A gyroplane is illustrated in Fig. 5. The gyroplane rotor, although it is distinct mechanically from the autogiro, possesses so many similarities in an aerodynamic sense that it is difficult to discuss the two separately. Both types of rotors have the property of developing their maximum lift-drag ratios at extremely low lift coefficients, and both develop about the same value of the ratio. Both rotors have the same possibility for a large speed range and for high-speed flight, although, as was the case with the autogiro, these possibilities have not been realized as yet by the gyroplane. The aerodynamic similarity of the two systems is, in fact, so close as to be almost an identity, and one is almost justified in saying

Fig. 4—The Direct-Control Autogiro C 30-P

that the only essential difference between the two rotors is a structural one. It is probable that, because the gyroplane blades must carry bending moments to the hub, the gyroplane rotor will be slightly heavier than the autogiro. No particular advantage in simplicity appears to exist for either rotor. The N.A.C.A., having examined closely the possibilities of the gyroplane, is carrying forward at present a program of wind-tunnel research on this system which includes the testing of a 10-ft. diameter model of the rotor. The information to be obtained from this test is expected to be the starting point of a further program of research on the rotor system.

Considering the gyroplane as it will probably be developed in the near future, the following statements are arrived at in the criterion:

1. The low-speed control is superior to that of the airplane.

2. The reliability is equivalent to that of an airplane, and emergency landings will be easier.

(Continued on page 131)

Fig. 5—The Wilford Gyroplane

Ideal Transmission Performance Set As Criterion for Development

By E. S. Hall

DRIVING an automobile is easy enough when you know how, but the present system of automobile control is unnecessarily complex and difficult to learn. It is remarkable chiefly in the degree of refinement which has been attained in making a crude and indirect system practical in the hands of the public. It has never been a logical system because it was not worked out from a functional basis. Instead of providing a system for controlling the automobile, means were provided for handling the engine, clutch, and transmission.

Without reference to these component parts, what would be a rational system of controlling the *automobile?* What would be the simplest and easiest automobile control system?

Offhand the simplest control system would seem to be that having the fewest pedals and levers, but the essential objective is not the simplest control system, but simplicity in control. Easy and simple control is not to be gained by concentrating a multiplicity of functions in a single lever or pedal; in that direction lies confusion. The simplest system to control is that in which each control agency has only one function and is operable independently from all the others in controlling that function.

The essential control functions are easily outlined. It is necessary to steer, to determine whether the car will go forward or backward, and to control the speed. In other words, control of an automobile is a matter of controlling the direction, the sign, and the magnitude, of its velocity.

Directional control is taken care of by the steering wheel, which is an eminently satisfactory type of control device. It has only one function, though it works both ways. It operates progressively, and can be judged as to extent by the immediate results.

Control of the sign of the velocity may be managed by a single agency, producing forward drive at one extreme of its movement and reverse at the other, with a disconnected or neutral point midway between. The present gear shift lever might be retained for this purpose, but it is desirable that both hands be free for steering, not only while driving on the open road, but also when turning around or parking in close quarters. It is preferable, therefore, that the fore and aft control be a single pedal of the rocker type, in the place now occupied by the clutch pedal.

Control of the magnitude of the velocity is a matter of acceleration and deceleration, and the latter is already taken care of by the brake pedal in a reasonable way. Control of acceleration, however, is now ridiculously complex, requiring the manipulation of the accelerator, clutch pedal and other clutch controls, and the gear shift lever. Car acceleration should be controlled entirely by the accelerator, and the control should be such that the driver could get more speed, up to the maximum attainable by the full power of the engine, merely by "stepping on it." The rapidity with which the car accelerates should correspond to the rapidity of movement of the pedal.

The complete control system, as shown in Fig. 1, consists of the steering wheel and a pair of rocker treadles with the brake pedal placed symmetrically between them. The brake pedal might also be a rocker if the brakes are power operated. Both the gear shift and hand brake levers have been eliminated. Instead of a separate lever for applying the brakes when parking, the convenient control on the steering column may be released to set the dog which holds the brake pedal in position with the brakes engaged.

Any proposed revision of the conventional control system should be consistent with present driving habits. In changing to the new standard control system described, gear shifting may be forgotten, but no change in habit is required in operating the brake and accelerator pedals, and the forward and reverse control is essentially a two-way clutch pedal. In driving, the feet would rest comfortably on the two rockers, controlling the sign of the velocity with the left, and the magnitude with the right, with both hands free to handle the direction. More speed could be had under any conditions by simply depressing the accelerator. Less speed could be had by letting up on the accelerator and depressing the brake as usual.

[This paper was presented at a meeting of the Metropolitan Section, Dec. 21, 1933.]

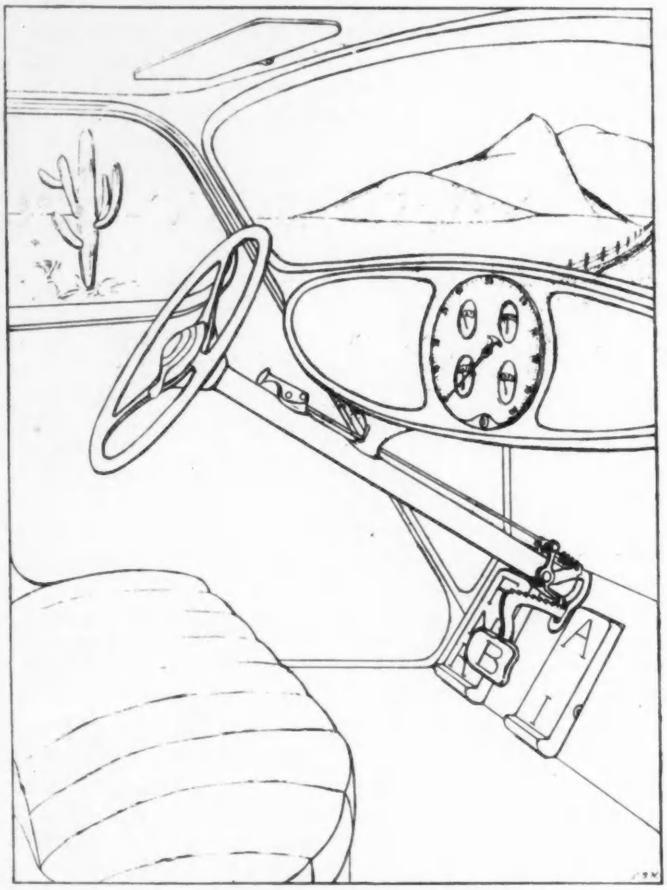


Fig. 1—New Standard Control System

The left rocker controls forward, neutral, and reverse; the right, all phases of car acceleration. The central brake pedal could also be a rocker. The convenient "over-center" control on the steering column permits the brakes to be locked in engagement for parking.

Braking with the engine does not belong in this picture[1], but equivalent braking ability must be provided.

Objections may be raised that the new standard control system would take away "the thrill of mastery" which comes from learning how to handle an illogical and mysterious bunch of levers and pedals. "Maybe the public likes to shift gears." If so, why make the engine big enough to do everything in high?

Objections to the new standard system of control may also be raised on the grounds that it is "academic," but it certainly can be realized in practice by finding the right solution for the transmission problem.

The Approach to the Transmission Problem

The most obvious way to approach the problem is to start with what we now have, and provide automatic means for controlling the clutch and gear shift. Possibly self-shifting gear boxes may be used for a few years, but it is hard to sustain any enthusiasm for them. They don't do enough. They make no fundamental attack on the transmission problem. The dissatisfaction with the present solution goes deeper than can be reached by any such superficial treatment as removing the shift lever. It is rooted in the fixed-ratio nature of the gear box.

While it is true that three or four "speeds" are provided, only one can be used at a time. In any fixed ratio, the engine is compelled to operate over a wide speed range. Maximum power cannot be delivered over this entire range, but only at one point toward the upper end of it. Optimum economy can only be had by running at speeds toward the lower end of the speed range. Obviously with any fixed ratio device, it is impossible either to make use of the full power of the engine or to operate economically, except momentarily, and there is never any possibility of having both optimum economy and maximum accelerative ability.

This basic defect of the gear box is illustrated at its worst in this country where the gear box is essentially a one-ratio device. Nobody wants to drive in anything but top gear. To gain this end, an overgrown engine is provided and the final drive gear ratio is chosen to give smooth high gear performance from 8 to 80 mi./hr., requiring the engine to run smoothly from 400 to 4000 rev./min. Consequently the high gear accelerative ability at slow car speeds is poor, the engine speed is excessive at high car speeds, and the gasoline mileage is unnecessarily and inexcusably low.

For use with the present type of internal combustion automobile engine, the ideal solution of the transmission problem lies with some form of transmission by means of which one shaft may drive another at any ratio continuously variable under load between desired limits, a decrease in the speed of one shaft being accompanied by a corresponding increase in its torque relative to the other shaft. In other words, the right solution must be a practical, efficient, quiet, and inexpensive continuously variable transmission, properly controlled.

Present State of the Art

Attempts to solve the continuously-variable transmission problem for automobiles have been made from many different directions. The following brief summary makes no pretense at completeness or finality, but indicates the range of the work that has been done, with comments as to the apparent desirability of each class.

Slipping clutches will not fill the bill. Some of the centrifugal clutches, with free-wheeling and a big enough engine, are exceptionally convenient to drive, and would almost tempt one to ask, "What more can anyone want?" But they haven't the accelerating punch without shifting gears; they are not fool-proof unless their automatic action is locked out whenever free-wheeling is locked out; and they cannot improve fuel economy, being wasters of power when slipping. They do not simplify the control system, for the clutch pedal must be retained and additional means provided for changing from automatic to pedal control. When fully engaged, they are no different from any engaged clutch, and in no sense do they cure the basic defect of the gear box.

The hydraulic clutches which consist of some form of positive displacement oil pump with means for throttling the output, have been misnamed "transmissions" in some instances, but are only slipping clutches, usually with plenty of slip. They are not solutions of the transmission problem.

[1] The use of the engine as a brake is justifiable only on the grounds that the brakes are inadequate. The notion that "you go down-hill on compression" is erroneous. The braking ability of the engine is due only to pumping and friction losses, the latter made more harmful by dumb carburetor design which provides no alternative to pouring the rich idling mixture through the engine when braking, resulting in damage to piston and cylinder lubrication, especially when descending long grades in mountainous country. All fuel fed through the engine when used for braking is worse than wasted.

Bearing loads induced by using the engine as a brake in the lower gears, especially in trucks, may exceed any encountered in normal operation, due to the excessive inertia forces of abnormal engine speed.

The engine is a power device; to use it as brake lining ought not to be necessary or possible.

The same may be said of the turbine clutches or "fluid flywheels." They may be smooth and delightful to drive, but they are only clutches or hydraulic couplings, and require supplementary change gearing. They do not avoid the basic defect of the gear box.

The turbine transmissions with a stationary reaction member can amplify torque, and there may be some hope of solving the transmission problem with them if their operating characteristics, efficiency, and overall production cost including the reverse gear, can be made satisfactory.

Considerable experience has been had with hydraulic transmissions which use some sort of variable displacement fluid pump to drive a fluid motor. Both those using cylinders parallel with the shaft, and those using radial cylinders in both pump and motor, have had success in marine and industrial applications, and many other types of pumps have been proposed. As an automobile transmission, it has not been easy to avoid too much bulk without running into excessive working pressures. Other difficulties include fluid friction losses, leakage, cavitation of the fluid, overheating, and too much weight and cost. The system of squeezing oil at high pressure out of one collapsible chamber into another to make the latter expand, is attractive on paper, but in spite of repeated attempts, has had no commercial success as an automobile transmission.

The pneumatic transmissions, using compressed air in a closed circuit, are similar in principle and in difficulties to the hydraulic. As applied to motor vehicles, the heat of the engine exhaust can be used to increase the efficiency of the system by heating the air compressed by the engine-driven compressor, before passing it to the air motor.

The electric drive is successfully used in buses and locomotives, but is so far relatively heavy and expensive. If these difficulties could be avoided, it might become acceptable for automobiles.

Mechanical continuously variable transmissions have been either friction or ratchet drives. The only one which does not fall in these classes is the "PIV" gear made by the Link-Belt Co. While an excellent device for its purpose, it does not appear to be suitable for automobiles.

Many varieties of friction drives have been proposed. Besides the familiar flat plate and friction ring type used in early automobiles, there are the parallel cone types, the spherical types, and the toric transmissions. Interest in the toric transmissions especially is now quite active in the automobile industry both here and abroad.

The ratchet drives comprise some means for providing reciprocation or oscillation which may be varied in extent, with means comprising one-way clutches associated with the driven member, for sorting out the positive from the negative impulses. Many ways of providing the variable impulses have been proposed. An elementary type uses rocking bars, rocked by operable connection with fixed cranks on the driving shaft, and connected to one-way clutches on the driven shaft by connecting rods which can be shifted along the rocking bars to vary the length of their strokes. In another system, variable eccentric cranks have been used; in another, a swash plate with a variable tilt. An intriguing method is that using a planetary system in which "rotating weights moving on epicyclic trajectories produce positive and negative impulses by virtue of centrifugal force." Various methods of interfering with the free vibrations of vibrating masses have been used to obtain the variable impulses in the so-called "inertia" types.

Excellent demonstrations have been made with a number of ratchet transmissions, but in some of them the possibility for adequate control appears to be problematical, others are troubled by noise due to backlash, and none of them are any better than their one-way clutches.

The ideal transmission may come from any of these classes or from an entirely different direction. Whatever its construction, the duty of a continuously variable transmission, and of its automatic and driver-operable controls, can profitably be explored now.

What Should the Transmission Do?

The term transmission, in its broad sense, includes all the mechanism by which the engine power is transmitted to the wheels which drive the car. Any discussion of the duty of the transmission involves the "overall gear ratio," an awkward phrase, and ambiguous since it is commonly used to denote both itself and its reciprocal. Rather than attempt to specify how the term "gear ratio" should be used, it is better to avoid confusion by not using it.

The reciprocal terms "rev ratio" and "drive ratio" may be used, defined as follows:

$$\text{Rev Ratio} = \frac{\text{Engine rev./min.}}{\text{Wheel rev./min.}}$$

$$\text{Drive Ratio} = \frac{\text{Wheel rev./min.}}{\text{Engine rev./min.}}$$

Note that in changing to what is commonly called a "lower gear," the rev ratio is raised to a higher numerical value, while the drive ratio is lowered. In discussing continuously variable transmissions, the "drive ratio" as above defined is usually the convenient term to use.

The following relationship is also of interest:

$$\text{Torque Ratio} = \frac{\text{Engine Torque}}{\text{Wheel Torque}} = \frac{\text{Drive Ratio}}{\text{Transmission Efficiency}}$$

The transmission should deliver the engine power to the driving wheels efficiently and in a quiet and fully acceptable manner at any drive ratio within desired limits, and with no interruption in the flow of power while the drive ratio is changing. Control should be so arranged that any driver will get maximum fuel mileage, yet with the full power of the engine always available on demand for acceleration and top speed. The right solution of the transmission problem demands not only a satisfactory continuously variable transmission, but also the proper automatic and driver-operable controls for it.

It is well known in general that maximum gasoline mileage is to be had by running the engine at the lowest speed at which the required power can be smoothly delivered. Maximum acceleration, however, can be had by running the engine at wide open throttle at the speed of peak horsepower. Maximum economy is always desirable; maximum acceleration is needed only occasionally, at the will of the driver. It is logical, therefore, to arrange the automatic control of the drive ratio to produce maximum economy, while the driver-operable control is arranged to over-ride the automatic control to produce more acceleration, up to the maximum, on demand.

To work out such a control system, answers to the following questions must be known:

What drive ratios are desirable for maximum economy consistent with smooth performance?

What drive ratios are desirable for maximum power applied at the wheels, for acceleration, hill-climbing, and top speed?

What is the duty of the automatic control in maintaining the right drive ratios for maximum economy?

What is the duty of the driver-operable control in modifying the automatic control when more power is wanted?

To answer these questions, it is necessary to estimate car performance. The 1933 sedan may be taken as a basis for comparison, but to get a fair idea of the total improvement attainable in acceleration and economy, the streamlined car which will have evolved by the time any new transmission system can be developed, should be considered.

Four cases, involving these two cars, are of particular interest, and will serve to illustrate the analytical method, as follows:

Car 33
- I 80 hp. 1933 sedan with the usual gear box.
- II 80 hp. 1933 sedan with a continuously-variable transmission.

Car 37
- III 80 hp. 1937 streamlined sedan with continuously-variable transmission.
- IV 40 hp. 1937 streamlined sedan with continuously-variable transmission.

The following specifications are common to both cars, except as noted:

Overall Length 174 in.; Width 68 in.; Height_{33} 68 in.; Height_{37} 64 in.; Wheelbase 114 in.; Tread 56 in.; Road Clearance 9 in.; Tires 6.00-17 in.; Rolling Radius of Wheel 14 in.

$$\frac{\text{Driving Wheel rev./min.}}{\text{Car mi./hr.}} = \frac{5280 \times 12}{60 \times 2\pi\, 14} = 12$$

Frontal Area, $A_{33} = 24$ sq. ft. $A_{37} = 23$ sq. ft.

Weight_{33} with driver 2700 lb.; with five passengers 3300 lb. $= W_{33}$.

Weight_{37} with driver 2250 lb.; with six passengers 3000 lb. $= W_{37}$.

[2] This sketch and the accompanying description were made before the author was aware of the extent to which streamlining was being advanced by Chrysler and others for 1934.

The gear box ratios used with Car 33 and the resulting values of the "Speed Ratio" $= \dfrac{\text{Engine rev./min.}}{\text{Car mi./hr.}}$ are as follows:

	Transmission Reduction		FinalDrive Reduction		Wheel rev./min. / Car mi./hr.		Speed Ratio
High Gear	1	×	4.17	×	12	=	50
2nd Gear	1.6	×	4.17	×	12	=	80
Low Gear	2.8	×	4.17	×	12	=	140

Car 33 need not be illustrated as its appearance is familiar. The performance of Car 33 will be calculated as a check on the accuracy of the assumptions and analytical method, the following known performance being recorded in advance:

Top Speed 80 mi./hr.

Acceleration from 0 to 65 mi./hr. through the gears in 23 sec.; average rate of acceleration, 2.8 mi./hr./sec.

Acceleration from 7 to 65 mi./hr. in high gear in 25 sec.; average rate of acceleration, 2.3 mi./hr./sec.

Gasoline Economy 19 mi./gal.

Car 37 is illustrated in Fig. 2[2]. Proportions are much the same as for the smaller cars of today, with no more overhang beyond the axles either front or rear. Passengers sit amidships, and both seats are wider than usual, with room for three passengers in each. Upholstery may be without springs or padding, cellular air cushions being used to save weight and increase comfort. The construction is such that the cushions may be easily removed for use outside the car, or rearranged inside the car to suit individual comfort, or for reclining while travelling, or for sleeping in camp.

Windows are big enough to see out of. The driver can see the road within ten feet of the front wheels and an overhead traffic light without using a periscope. The design objective was visibility, not economy in the use of safety glass.

Headroom is 4 to 10 in. more than now provided. With 9 in. road clearance, the floor is normally 12 in. above the road, and the ceiling is 51 in. above the floor. The additional headroom is obtained by incorporating the chassis frame in

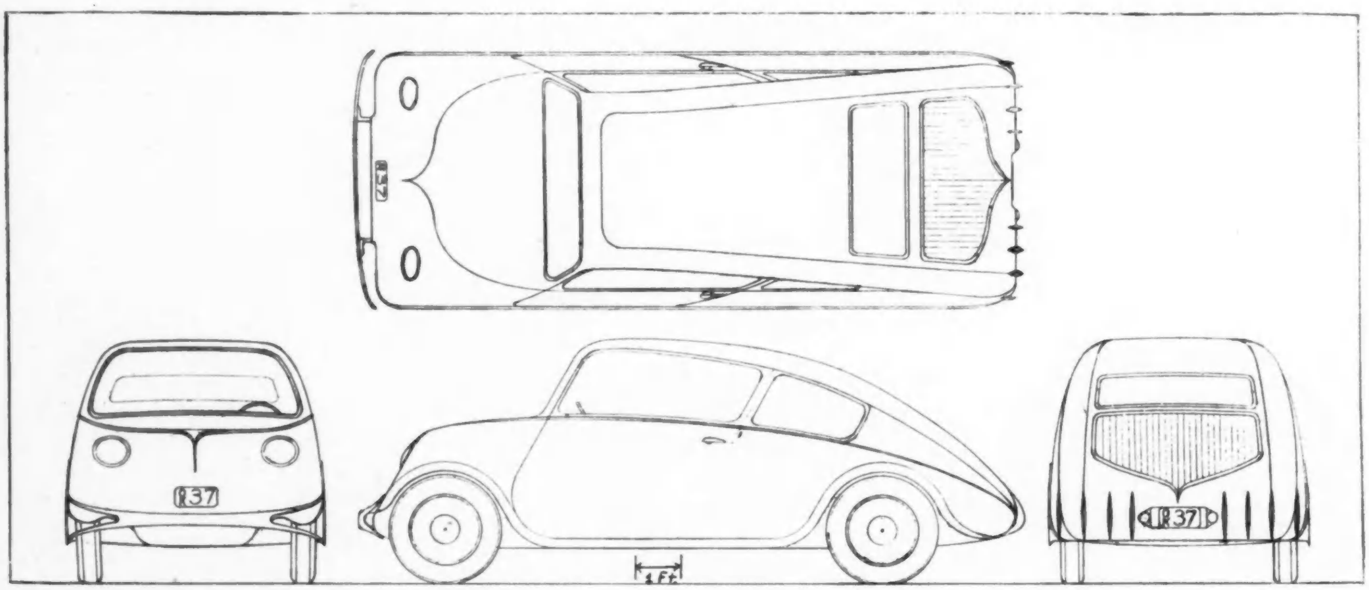


Fig. 2—Car 37, an example of reasonably clean streamlining both above and below, with much more headroom and visibility than provided in any 1933 cars.

TABLE 1

	Resistance—Lb				Required Horsepower							
	Rolling Resistance		Air Resistance		Rolling Resistance Horsepower		Air Resistance Horsepower		Car Horsepower		Brake Horsepower	
mi./hr.	33	37	33	37	33	37	33	37	33	37	33	37
10	41.25	37.5	3.60	1.38	1.1	1	0.096	0.037	1.196	1.037	1.50	1.30
20	41.25	37.5	14.40	5.52	2.2	2	0.768	0.294	2.968	2.294	3.71	2.87
30	41.25	37.5	32.4	12.4	3.3	3	2.59	0.993	5.89	3.993	7.36	4.99
40	41.25	37.5	57.6	22.2	4.4	4	6.15	2.36	10.55	6.36	13.2	7.95
50	41.25	37.5	90.0	34.5	5.5	5	12.01	4.60	17.51	9.60	21.9	12.0
60	41.25	37.5	129.6	49.7	6.6	6	20.75	7.94	27.35	13.94	34.2	17.4
70	41.25	37.5	176.4	67.6	7.7	7	32.95	12.61	40.65	19.61	50.8	24.5
80	41.25	37.5	230.4	88.4	8.8	8	49.2	18.83	58.0	26.83	72.5	33.6
90	41.25	37.5	291.6	111.8	9.9	9	70.1	26.8	80.0	35.8	100.0	44.7
100	41.25	37.5	360	138	11.0	10	96.0	36.8	107.0	46.8	134	58.5
110	41.25	37.5	436	167	12.1	11	128.0	48.9	140.1	59.9	175	74.9
120	41.25	37.5	518	199	13.2	12	166	63.5	179.2	75.5	224	94.4

the body structure, and by the transfer of the engine to the rear. The engine, transmission and rear wheels comprise a compact power and driving unit which is easily removable for servicing or for use with another body and front wheel assembly. The front wheels are independently sprung.

The doors are large and the seats not too low, so that with the unusually high ceiling, it is easy to get in and out without using a shoe horn. Old folks will find it a blessed relief after the doubling-up process they have been forced to comply with to suit immature ideas of speedy appearance. Yet actual speedy appearance remains, and the car is well streamlined, even the under surface next to the road being clean and smooth. However, streamlining is confined to the elevational aspect; there is little of it in the plan view. The full plan of the car is therefore available for useful purposes, and side winds have no tendency to produce transverse "lift" to move the car off the road.

While there is more room in the body than in today's sedan, additional storage space is available "under the hood." Within this storage compartment the spare wheel is easily stowed, occupying the space left vacant by the removal of the radiator to the rear. Steam heat may be piped where necessary for comfortable winter driving, without any necessity for either engine or radiator being at the front.

Estimating Performance

The performance of Cars 33 and 37 may now be estimated according to the following procedure:

1—From the specifications of the car, estimate the required horsepower.

2—Select an engine with peak horsepower equal to the required horsepower at the desired top speed.

3—Estimate and plot the acceleration obtainable with the maximum horsepower of the engine applied at the optimum drive ratio.

4—Estimate and plot the gasoline mileage as a function of car speed, for selected engine speeds, and for selected engine load factors.

5—Plot the useful rev ratios and/or drive ratios as functions of car speed, for selected engine speeds and for selected engine load factors.

6—From a study of the foregoing, and from practical considerations including the necessity for an infinite rev ratio (= zero drive ratio) at idling speed, and the probable necessity for temporarily adjusting the idling speed to a higher value when the engine is cold, determine the desired system of operation, and outline the duty of the automatic control, and of the driver-operable control.

1. *Required Horsepower.*—The usual method of estimating and plotting the required level road horsepower as a function of car speed may be used. For smooth pavement, assume:

$$\text{Rolling Resistance} = 0.0125 \text{ lb./lb. of gross car weight}$$
$$RR_{33} = 0.0125 \times 3300 = 41.25 \text{ lb.}$$
$$RR_{37} = 0.0125 \times 3000 = 37.5 \text{ lb.}$$

By definition,

$$HP = \frac{(\text{lb.}) \times (\text{ft./min.})}{33{,}000} = \frac{(\text{lb.}) \times 88\ (\text{mi./hr.})}{33{,}000}$$
$$= \frac{(\text{lb.})}{375}(\text{mi./hr.})$$

Rolling Resistance HP

$$RHP_{33} = \frac{41.25\ (\text{mi./hr.})}{375} = 0.11\ (\text{mi./hr.})$$

$$RHP_{37} = \frac{37.5\ (\text{mi./hr.})}{375} = 0.1\ (\text{mi./hr.})$$

$$\text{Air Resistance} = K\,A\ (\text{mi./hr.})^2 \text{ lb.}$$

Assume: $K_{33} = 0.0015$

and $K_{37} = 0.0006$

Then $AR_{33} = 0.0015 \times 24\ (\text{mi./hr.})^2$ lb.

and $AR_{37} = 0.006 \times 23\ (\text{mi./hr.})^2$ lb.

Air Resistance HP

$$AHP_{33} = \frac{0.0015 \times 24}{375}\ (\text{mi./hr.})^3$$

$$AHP_{37} = \frac{0.0006 \times 23}{375}\ (\text{mi./hr.})^3$$

In each case, HP Required at the Wheels,

$$CHP = RHP + AHP$$

Assuming 80 per cent efficiency of transmission to the wheels,

$$\text{Engine Hp. Required, } BHP = \frac{CHP}{0.80}$$

Values from these equations are tabulated in Table 1 and the curves are plotted in Fig. 3.

The assumed values of the rolling resistance and of the coefficient K in the air resistance equation are reasonable

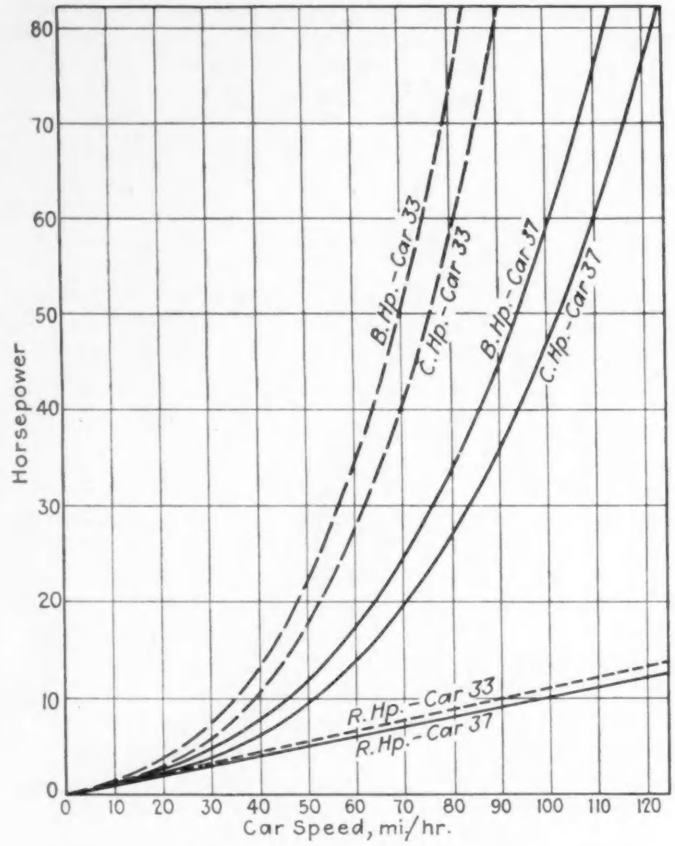


Fig. 3—Rolling Resistance Horsepower = R.Hp., Horsepower Required at the Wheels = C.Hp., and Engine Horsepower Required = B.Hp., for Cars 33 and 37

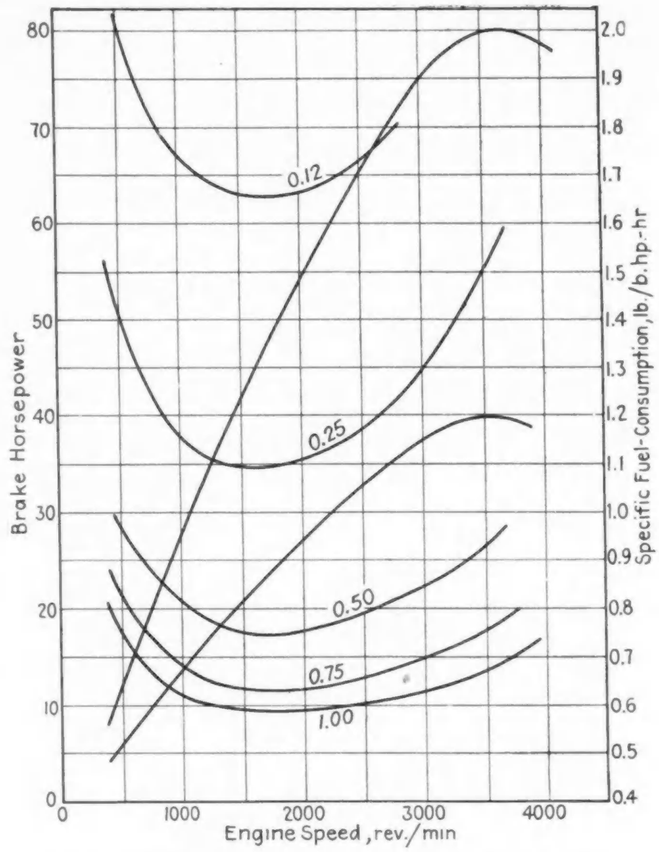


Fig. 4—Maximum Horsepower and Fuel Consumptions at Various Loads of the Two Engines Considered

according to various authorities.[3] The trend toward lower pressure tires may demand increase in the assumed rolling resistance, and it is also probable that rolling resistance increases somewhat with speed. On the other hand, the transmission losses assumed are relatively high. A transmission efficiency of 80 per cent is exceeded in the gear box, and will probably be exceeded in all ratios by any acceptable continuously variable transmission.

Before jumping to the conclusion that K_{37} is too low, Car 37 should be studied enough to realize that, except for sacrifice at the windshield for the sake of a far wider range of visibility than is now provided in any production car, it is a clean job of streamlining, top, sides, and underneath, with practically no "wind claws".

For every different car and engine to which a continuously variable transmission is to be applied, this problem must be reworked. The examples here illustrate the analytical method, and present a fair picture of the practical limits toward which improvement in performance may proceed. Those who feel that the assumptions are too optimistic may substitute another 1937 sedan for which the assumed values will be acceptable.

2. *Selection of the Engine.*—When choosing an engine for use with a continuously variable transmission, level road conditions only need be considered. Hills or headwinds will cut the maximum speed, but with the full engine power always available at the optimum rev ratio, the final drive gear ratio is of little significance. Acceleration and hill climbing ability for a car of given weight will depend only on the maximum power of the engine and the traction available. The preliminary choice of the engine can be based on peak horsepower equal to the required horsepower at the desired top speed.

Car 33 comes equipped with an 80 hp. engine having the power curve shown in Fig. 4. To insure sound conclusions from this analysis, the engine curves started with should be representative. The 80 hp. curve of Fig. 4 was therefore checked against the power curves of two other engines of the same size. For convenience, the power curve of the 40 hp. engine was taken as half that of the 80 hp. engine at each speed, the validity of this assumption being checked by comparison with the actual horsepower curves of two engines of about 40 hp. The specific fuel consumption curves are composites from part load data on five different engines, and can be applied fairly to both the 80 hp. and the 40 hp. engines considered.

From Figs. 3 and 4 it appears that the estimated top speed of Car 33 with the gear box is 82 mi./hr. the engine delivering 78 b.hp. at 4100 rev./min. This fairly checks known performance. With a continuously variable transmission, the engine speed would be held at 3500 rev./min. delivering 80 b.hp., and the top speed would be slightly faster. Car 37 with an 80 hp. engine would do 112 mi./hr., and with an engine of only 36 hp., would have the same top speed as the 80 hp. Car 33, under the road and air conditions assumed.

3. *Accelerative Ability.*—The accelerative ability can be

[3] Engine and Car Performance by W. S. James; see S.A.E. TRANSACTIONS, Vol. 23. Practical Tractive Ability Methods by A. M. Wolf, and all papers listed in the bibliography therein; see S.A.E. JOURNAL, December, 1930. Is 50 Miles Per Gallon Possible with Correct Streamlining? by W. E. Lay; see S.A.E. JOURNAL, April and May, 1933. Alexander Klemin, Letter, December 18, 1933.

worked out and plotted as a function of car speed, in connection with the tractive effort which is the force applied to the road by the wheels. The tractive effort may be found from the wheel torque, but is more easily computed directly from the horsepower equation:

$$HP = \frac{(\text{lb.}) \times (\text{ft./min.})}{33{,}000}$$

which in this case would read:

$$CHP = \frac{TE \times 88\ (\text{mi./hr.})}{33{,}000}$$

Substituting the equivalent engine horsepower, according to the assumed transmission efficiency of 80 per cent

$$0.80\,BHP = \frac{TE}{375}\ (\text{mi./hr.})$$

or
$$TE = \frac{300\ (\text{b.hp.})}{(\text{mi./hr.})}\ \text{lb.}$$

Results from the application of this equation to Cars 33 and 37 are in Table 2. In order to find the maximum tractive effort in each gear ratio for case 1, the engine speed at each car speed must be known. This is easily figured from the respective "speed ratio". Then the corresponding maximum brake horsepower can be read from Fig. 4, and the values of the tractive effort figured from the above equation.

In figuring the tractive effort with a continuously variable transmission, the maximum brake horsepower only need be considered because the full power of the engine can be available at any car speed by operating at the drive ratio that permits the engine to run at the speed of peak power. When the quantity, (b. hp.) is a constant parameter, the equation:

$$TE = \frac{300\ (\text{b.hp.})}{(\text{mi./hr.})}\ \text{lb.}$$

is in the form $y = \frac{a}{x}$ and the curves are equilateral hyperbolas.

These curves for the two engines are plotted in Fig. 5, together with the three TE curves for the 80 hp. engine with the gear box. In each case, the tractive effort plotted is the maximum available and includes that required to maintain the car speed. In order to find the tractive effort available for acceleration, TE_a, the amount required to maintain speed must be subtracted. This required tractive effort is equal to the sum of the rolling resistance plus the air resistance. The subtraction can be done graphically by plotting the required tractive effort on Fig. 5. This curve also represents zero accelerative ability, and if a series of similar curves be drawn having initial ordinates corresponding to a suitable accelerative ability scale, the accelerative ability can be read directly.

To determine the desired scales, the accelerative ability is proportional to the tractive effort in accordance with the relationship:

$$f = m\,a$$

or
$$a = \frac{TE_a}{\text{mass of the car}}\ \text{ft./sec./sec.}$$

$$= \frac{TE_a \times 32.2 \times 3600}{W \times 5280}\ \text{mi./hr./sec.}$$

With car weights of 3300 lb. and 3000 lb. respectively,

$$a_{33} = \frac{TE_a}{150}\ \text{mi./hr./sec.} \qquad \text{and} \qquad a_{37} = \frac{TE_a}{136.4}\ \text{mi./hr./sec.}$$

TABLE 2

	Tractive Effort—Lb.											
	Required TE = RR+AR		Maximum Tractive Effort from 80 Horsepower Engine With Gearbox and Wide Open Throttle								Maximum Tractive Effort with a Continuously Variable Transmission from:	
			Direct Drive $\frac{\text{rev./min.}}{\text{mi./hr.}} = 50$			2nd Gear $\frac{\text{rev./min.}}{\text{mi./hr.}} = 80$			Low Gear $\frac{\text{rev./min.}}{\text{mi./hr.}} = 140$				
mi./hr.	33	37	rev./min.	Brake Horse-power from Fig. 4	Tractive Effort	rev./min.	Brake Horse-power from Fig. 4	Tractive Effort	rev./min.	Brake Horse-power from Fig. 4	Tractive Effort	80 Horse-power Engine	40 Horse-power Engine
5						400	9	540	700	18	1080	4800	2400
10	44.8	38.9	500	12	360	800	22	660	1400	40	1200	2400	1200
20	55.6	43.0	1000	28	420	1600	45	675	2800	72	1079	1200	600
30	73.6	49.9	1500	42	420	2400	63	630	4200	77	770	800	400
40	98.8	59.7	2000	54	405	3200	77	578				600	300
50	131.2	72.0	2500	65	390	4000	79	474				480	240
60	170.8	87.2	3000	75	375							400	200
70	217.6	105.1	3500	80	343							343	171
80	272	126	4000	79	296							300	150
90	333	149										267	133
100	401	176										240	120
110	477	204										218	109
120	559	236										200	100

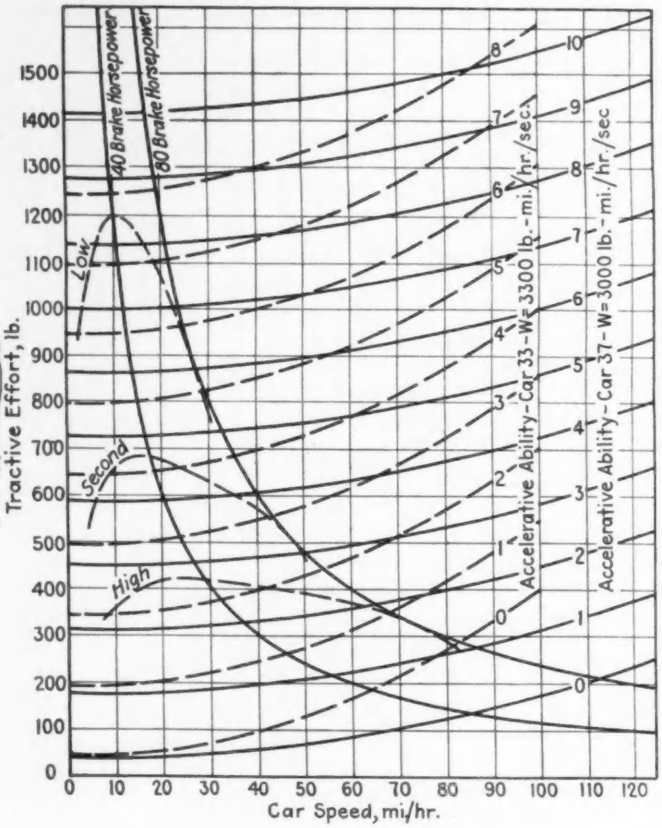


Fig. 5—Tractive Effort and Accelerative Ability

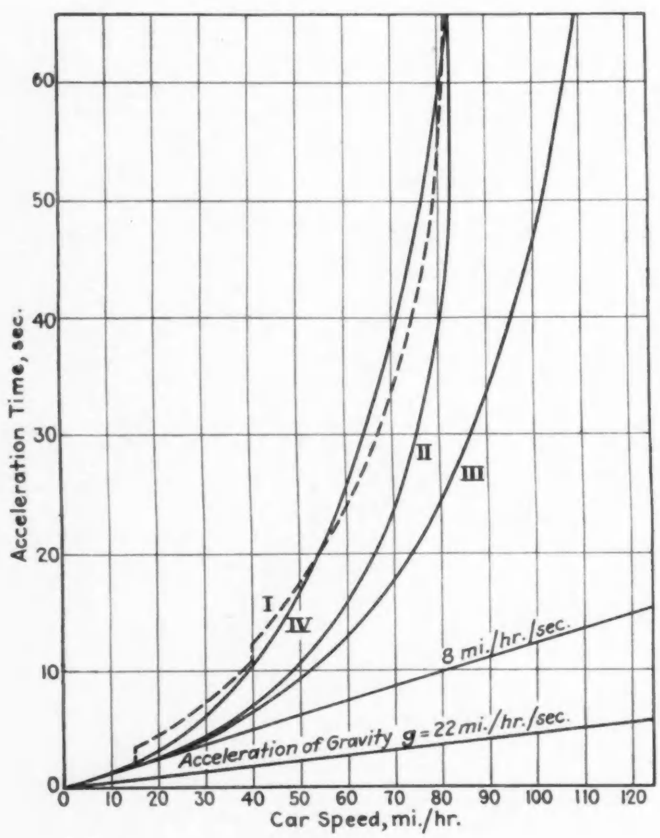


Fig. 6—Acceleration Time for Cars 33 and 37

In Fig. 5, the total tractive effort is to be read from the horizontal lines, but the accelerative ability must be read from the similar curves, of which there are two sets, with corresponding scales, the left-hand scale referring to the accelerative ability of Car 33, and the right-hand scale, to that of Car 37.

The penalty imposed by a fixed-ratio gear box is at once evident. The full power of the 80 hp. engine, when operating in any fixed gear, can be available for acceleration only at the moment when the corresponding *TE* curve is tangent to the 80 hp. hyperbola, and the maximum accelerative ability in each gear, low, second, and high, is approximately equal to that available at the same car speeds, respectively, from an engine of half the power with a continuously variable transmission.

The influence of car weight on acceleration is also brought out. For example, the maximum rate of acceleration possible in high gear with Car 33 is 2.4 mi./hr./sec., but if the car weighed 3000 lb. instead of 3300 lb., its maximum acceleration would be at the rate of 2.7 mi./hr./sec. Accelerative ability with a given amount of tractive effort available is inversely proportional to the weight of the car. It is better to see how light an engine will give enough acceleration, rather than how powerful an engine can be carried around.

The maximum acceleration possible, however, no matter how powerful the engine, is substantially independent of the weight of the car. With less weight on the driving wheels, a correspondingly lower tractive effort will cause the wheels to slip. The total tractive effort divided by the weight on the driving wheels, is the coefficient of friction:

$$\mu = \frac{TE}{W_d} = 0.6 \pm 0.4 \text{ depending on road conditions, when}$$

slipping is imminent. Assuming two-thirds of the car weight on the driving wheels, and average road conditions with a friction coefficient of 0.6, a tractive effort of $0.6 \times 2200 = 1320$ lb. would almost slip the wheels of Car 33, and of $0.6 \times 2000 = 1200$ lb. would almost slip the wheels of Car 37. These two values of the tractive effort correspond to the same accelerative ability of 8.4 mi./hr./sec. on each of the two scales. Obviously, under ordinary road conditions, the maximum rate of acceleration with two-wheel drive cannot much exceed 8 mi./hr./sec. with two-thirds of the weight on the driving wheels, or 6 mi./hr./sec. with half of it there. It appears doubtful if four-wheel drive could add enough more accelerative ability to be worth the additional cost and complication.

A point on Fig. 5 may represent a rate of accelerative ability at a given car speed. In Fig. 6, the elapsed time of acceleration is plotted against car speed, and the rate of accelerative ability is a slope of the curve at the point in question. Points on Fig. 5 may therefore be transferred to Fig. 6 as slopes, with the aid of a simple table of equivalent slope angles to suit the scales of Fig. 6.

In this manner, in Fig. 6, the acceleration time curves for each of the four cases under consideration have been plotted. The curve for Car 33 with the gear box, marked *I*, shows an acceleration from 0 to 65 mi./hr. through the gears in 28 seconds. The advertised accelerative ability of this car is 23 seconds as above stated. Presumably the advertised acceleration was obtained with the driver alone in the car, which would make the car weight 2700 lb. This analysis has been based on a five passenger load with the car

TABLE 3

High Gear Gasoline Mileage Car 33 with Gearbox

mi./hr.	rev./min. = 50 × (mi./hr.)	Brake Horsepower Used from Table 1 or Fig. 3	Brake Horsepower Available from Table 2 or Fig. 4	Load Factor $= \frac{B.Hp._u}{B.Hp._a}$	Lb./B.Hp. Hr. from Fig. 4	mi./gal.
10	500	1.50	12	0.125	2.2	18.9
20	1000	3.71	28	0.132	1.7	19.8
30	1500	7.36	42	0.175	1.4	18.2
40	2000	13.2	54	0.244	1.11	17.1
50	2500	21.9	65	0.337	1.0	14.3
60	3000	34.2	75	0.455	0.9	12.2
70	3500	50.8	80	0.635	0.83	10.4
80	4000	72.5	79	0.917	0.78	8.8

weight 3300 lb. Acceleration time is directly proportional to car weight. $\frac{2700}{3300} \times 28$ sec. $= 23$ sec.

Comparison of curves *I* and *II* in Fig. 6 shows that when the gear box of Car 33 is replaced with a continuously variable transmission, the rapidity of acceleration from 10 to 50 mi./hr. is increased about 70 per cent. Over the whole speed range, the improvement in accelerative ability due to the continuously variable transmission alone, would average about 40 per cent, the improvement being most marked below 60 mi./hr.

Streamlining has no appreciable effect on accelerative ability at speeds below 30 mi./hr., but from the way curves *II* and *III* diverge above 30 mi./hr., it is evident that the improvement due to streamlining increases rapidly as the car speed increases.

The most significant fact brought out by Fig. 6 is that Car 37 with a 40 hp. engine (*IV*) has approximately the same accelerative ability as today's Car 33 with twice the power (*I*).

4. *Gasoline Mileage.*—The gasoline mileage at any selected engine speed may be plotted as a function of the car speed, from the equation:

$$(\text{mi./gal.}) = \frac{(\text{lb./gal.} = 6.25) \times (\text{mi./hr.})}{(\text{lb./b.hp. hr.}) \times (\text{b.hp.})}$$

Of course, the values used together in this equation must be coincident.

The use of this equation with a gear box is illustrated in Table 3. The engine speeds corresponding to the car speeds are easily tabulated from the "speed ratio" for high gear. Then the brake horsepower used at each car speed is obtained from Fig. 3 or Table 1. From Fig. 4 or Table 2 the maximum brake horsepower which the engine can develop at each engine speed can be read, and from these two brake horsepower values, the load factor can be figured. Then, by interpolation, the specific fuel consumption can be obtained from Fig. 4. The results are plotted in the lower portion of Fig. 7. This curve checks the known gasoline mileage of Car 33 as now running on the roads.

Interpolation in reading fuel consumption values can be avoided in using this equation, for a car equipped with a continuously variable transmission, by the method illustrated in Table 4. For each engine speed, the engine load factors for which specific fuel consumption curves are available, are tabulated, and the corresponding fuel consumption entered in each case, the values being read from Fig. 4. Then for each engine speed, the full load brake horsepower is obtained from Fig. 4, and entered in the brake horsepower column. This brake horsepower value is multiplied by the other load factors, and the corresponding part load brake horsepower values are entered. Then for a given car, the car speed for each brake horsepower value can be read from Fig. 3. With this data all tabulated, the slide-rule can provide the mi./gal. values, and the curves for gasoline mileage as a function of car speed, for selected engine speeds, may be plotted as shown in Figs. 7 and 8.

Crossing each series of curves, another series showing gasoline mileage as a function of car speed, for selected engine load factors, may be plotted from the same points. These curves are shown by dotted lines. Together, the two sets of curves give a complete picture of the conditions which control gasoline mileage on level pavements, and clearly indicate that optimum mileage is to be had by holding the engine speed down to a minimum until the throttle is wide open, further increase in car speed being gained by increasing the engine speed up to the speed of peak horsepower.

These same conditions will produce low oil consumption. Gasoline economy may be taken as representative of overall operating economy.

The curves for full load and 75 per cent load are close together at the lower engine speeds, and with some engines analyzed, actually cross. If the engine gets rough at 600 rev./min. toward full load, the engine speed may be allowed to increase at 75 per cent load, with no appreciable change in economy. This condition may not hold for all engines, but has been common to all so far investigated.

Some engines show less difference in economy between the curves for 600 and for 1000 rev./min. This might be expected from Fig. 4. If the specific fuel consumption is relatively high at the lower engine speeds, the economy of the engine may be better at 1000 rev./min. toward the bottom of the fuel consumption curves, than at 600. Lubrication conditions in the engine may be better at the higher engine speeds, and all things considered, it may prove preferable to run somewhat less laboriously at 1000 rev./min. or a little faster, rather than at the slowest engine speed at which the engine will carry the load.

The comparison in Fig. 7 shows that the gasoline mileage can be improved about 40 per cent simply by replacing the gear box with a continuously variable transmission. In Figs.

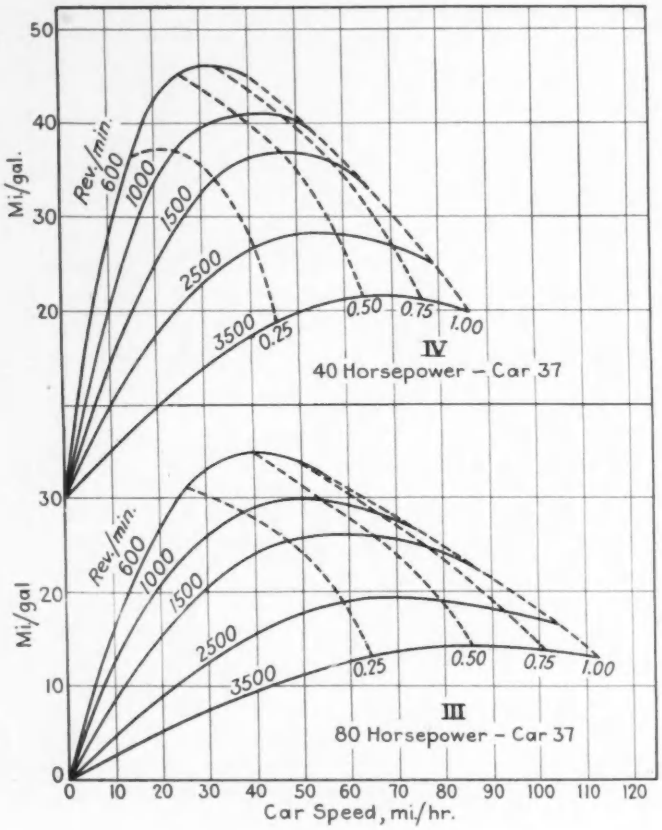


Fig. 8

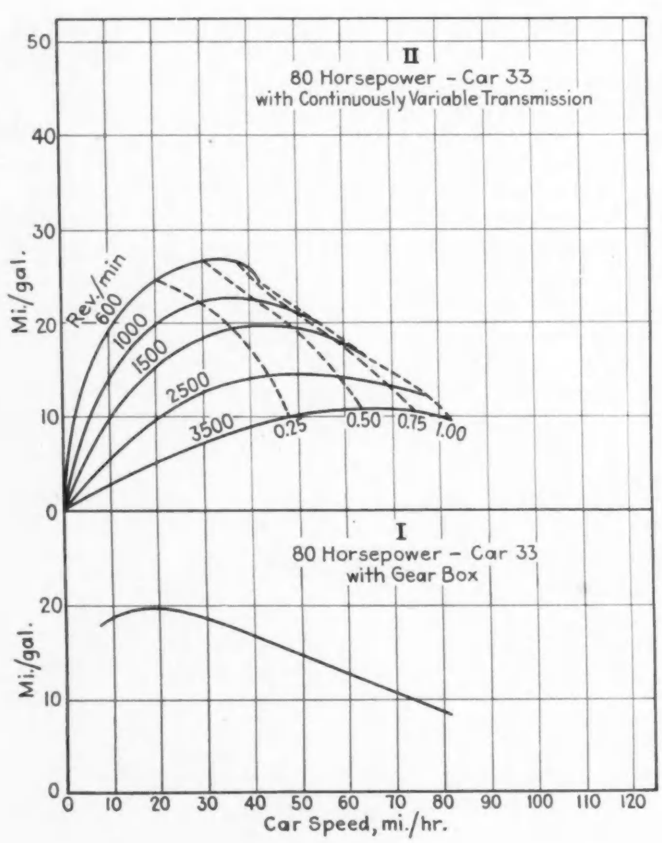


Fig. 7

Figs. 7 and 8—Gasoline Mileage

7 and 8, comparison of *II* with *III* shows that streamlining may improve fuel economy about 80 per cent more at cruising speeds. If at the same time that streamlining and the continuously variable transmission are adopted, full advantage be taken of both and the engine be reduced to a reasonable size, the gasoline mileage can be increased to about 40 mi./gal. on the average, as shown at *IV* in Fig. 8.

It is interesting to speculate on the effect of the continuously variable transmission on engine development. It puts a premium on economical engine operation at slow engine speeds. Flexibility and a high range of engine speed are no longer so important. Engine and carburetor development specifically for operation with a continuously variable transmission might possibly result in gasoline mileage averaging over 50 mi./gal., an increase of 200 per cent over the mileage now obtainable with a car of the same size and accelerative performance.

5. *Useful Transmission Ratios.*—All useful rev. ratios may be easily plotted as functions of car speed, for selected engine speeds, in accordance with the definition:

$$\text{Rev. Ratio} = \frac{\text{Engine rev./min.}}{\text{Wheel rev./min.}}$$

This is in the form $y = \frac{a}{x}$ and the resulting hyperbolas are applicable to all cars and engines, but the engine speeds selected as values of the parameter *a,* should include the minimum speed for smooth operation under load and the speed at peak horsepower for the engine under consideration.

If the engine speeds selected are the same as those used in plotting the gasoline mileage in Figs. 7 and 8, the values of car speed in mi./hr. can be taken from Table 4 and used to plot the load factor curves which cross the curves of constant engine speed in Fig. 9. All rev. ratios useful to maintain speed in still air on level pavements are included in these networks above the full load curves.

The hyperbolas of Fig. 9 could be plotted more conveniently on logarithmic paper where they would be straight lines at 45 deg. with the axes. A still better way to plot is to use the reciprocal relationship, plotting the drive ratios instead of the rev. ratios:

$$\text{Drive Ratio} = \frac{\text{Wheel rev./min.}}{\text{Engine rev./min.}}$$

For any constant engine speed, this is in the form $y = a\,x$ and the curves are straight lines radiating from the origin as shown in Fig. 10.

Figs. 9 and 10 are different pictures of the same thing. Fig. 10 is preferable because it is more easily drawn, because it has a useful similarity with Fig. 8, and because it contains the entire picture, while part of the curves of Fig. 9 extend to infinity and therefore cannot be shown.

While the curves for constant engine speed in Figs. 9 and 10 apply always, it is important to remember that the load factor curves can be used only to indicate the throttle opening that will maintain a given car speed for the specific car and engine, with the assumed air and road conditions. For example, referring to *IV* in either Fig. 9 or Fig. 10, with the engine at 600 rev./min., 25 per cent throttle opening will maintain a speed of 15 mi./hr. in still air on smooth and level pavement. Greater throttle opening at 15 mi./hr. would produce acceleration; less, deceleration.

6. *The Control of the Transmission.*—The operation of

TABLE 4

Gasoline Mileage with a Continuously Variable Transmission

rev./min.	Load Factor	Lb./B.Hp.Hr. from Fig. 4	80 Horsepower Engine					40 Horsepower Engine		
			Brake Horse-power (full load values from Fig. 4)	Car 33		Car 37		Brake Horse-power (full load values from Fig. 4)	Car 37	
				mi./hr. from Fig. 3	mi./gal.	mi./hr. from Fig. 3	mi./gal.		mi./hr. from Fig. 3	mi./gal.
600	1.00	0.72	15.2	42.3	24.2	56.0	32.0	7.6	39.5	45.2
	0.75	0.78	11.4	37.3	26.6	48.6	34.2	5.7	33.0	46.4
	0.50	0.93	7.6	30.1	26.6	39.5	35.0	3.8	25.6	45.2
	0.25	1.35	3.8	20.1	24.5	25.6	31.2	1.9	15.0	36.6
1000	1.00	0.62	28	55.1	19.9	74.2	26.8	14.0	54.0	38.9
	0.75	0.68	21	49.0	21.4	65.2	28.7	10.5	46.7	40.9
	0.50	0.81	14	40.9	22.6	54.0	29.8	7.0	37.2	41.0
	0.25	1.16	7	29.0	22.3	37.2	28.7	3.5	24.0	37.0
1500	1.00	0.59	42	64.7	16.3	87.7	22.1	21.0	65.2	32.9
	0.75	0.63	31.5	58.0	18.3	78.0	24.6	15.75	57.2	36.0
	0.50	0.75	21	49.0	19.5	65.2	25.9	10.5	46.7	37.0
	0.25	1.09	10.5	35.9	19.6	46.7	25.5	5.25	31.5	34.4
2500	1.00	0.60	65.2	77.1	12.3	104.6	16.7	32.6	79.0	25.2
	0.75	0.66	48.9	68.9	13.3	93.3	18.0	24.45	70.0	27.1
	0.50	0.79	32.6	58.8	14.3	79.0	19.2	16.3	58.1	28.2
	0.25	1.17	16.3	44.0	14.4	58.1	19.1	8.15	40.8	26.8
3500	1.00	0.67	80	82.7	9.7	112.5	13.1	40	86.0	20.0
	0.75	0.75	60	74.6	10.8	101.0	14.0	30	76.3	21.2
	0.50	0.93	40	63.6	10.7	86.0	14.4	20	63.8	21.5
	0.25	1.50	20	48.0	10.0	64.0	13.3	10	45.3	18.9

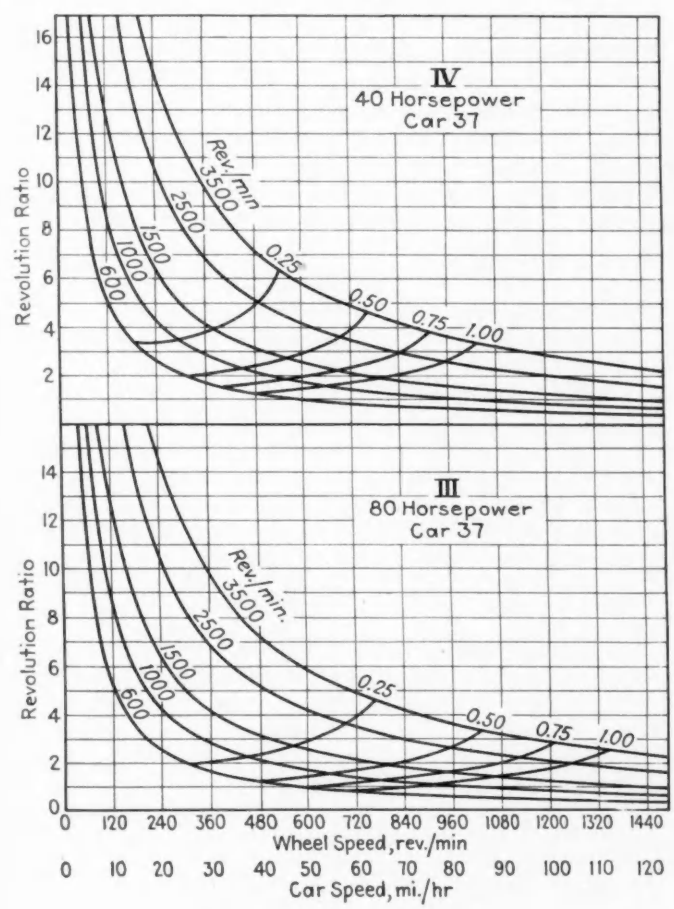


Fig. 9—Useful Rev. Ratios

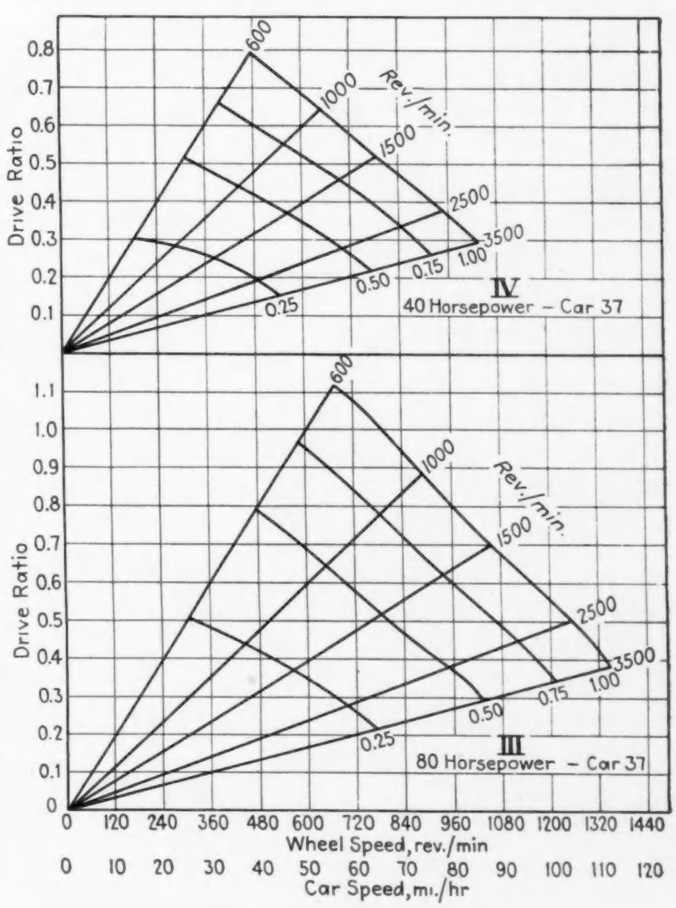


Fig. 10—Useful Drive Ratios

Table 5—Performance Comparison

Identification

I	Car 33 = 1933 Sedan, 80 Hp., Gear Box.
II	Car 33 = 1933 Sedan, 80 Hp., Continuously Variable Transmission.
III	Car 37 = Streamlined Sedan, 80 Hp., Continuously Variable Transmission.
IV	Car 37 = Streamlined Sedan, 40 Hp., Continuously Variable Transmission.

Top Speed			Comparative
I	82 mi./hr.		1.00
II	83 mi./hr.		1.01
III	112 mi./hr.		1.37
IV	86 mi./hr.		1.05
Accelerative Ability, 10 to 50 mi./hr.			
I	16.0 sec.	2.5 mi./hr./sec.	1.00
II	9.3 sec.	4.3 mi./hr./sec.	1.72
III	7.8 sec.	5.1 mi./hr./sec.	2.05
IV	15.4 sec.	2.6 mi./hr./sec.	1.04
Gasoline Mileage, Average Cruising 30-70 mi./hr.			
I	14.4 mi./gal.		1.00
II	20.7 mi./gal.		1.44
III	32.5 mi./gal.		2.25
IV	39.8 mi./gal.		2.75

the car with a continuously variable transmission and the new standard control system can now be outlined.

The engine can be started with the fore and aft control in neutral. To prevent the possibility of stalling the engine, the drive ratio should always be zero when the engine is idling.

For maximum economy, operation should be consistent with the top curves of Fig. 10. The car may be started by pushing the fore and aft control into forward position and advancing the accelerator slowly. As the engine speed is raised above the idling speed range, the drive ratio of the transmission should be raised from zero to that which will put load enough on the engine to hold it at uniform speed, gradually increasing the drive ratio as the car speed increases, until the throttle is fully opened. To get still more speed, the drive ratio must be decreased to permit the engine speed to increase so that the engine can develop more power.

If at any time, the driver desires more rapid acceleration than is available in the foregoing manner, more rapid advance of the accelerator should produce it. Maximum acceleration over the entire speed range is to be had by operation consistent with the bottom curves of Fig. 10.

Of course, free-wheeling is to be automatic whenever the accelerator pedal is returned to idling position. The only rational objections to free-wheeling have been due to: (1) the necessity for using the engine as a brake; and (2) the momentary delay in the application of power while the engine is speeding up, after free-wheeling.

The continuously variable transmission may wipe out both these objections. Part of the engineering job which must be done on the transmission system is to provide a fully adequate braking system to supplement the present wheel brakes. The desirable arrangement would be to construct the transmission so that rocking the fore and aft pedal into neutral, or even into reverse when driving forward and vice versa, would operate as an effective two-wheel braking system, independent of the four-wheel brakes. This would dispose of the first objection to free-wheeling.

The second objection may be less noticeable with a continuously variable transmission. Whenever the throttle is closed, the engine drops back to idling, the car free-wheels, and the drive ratio returns to zero, but when the throttle is opened again, the engine gets back into the collar at once, as soon as the engine speed has increased 200 rev./min., practically on the instant, and ample accelerative ability is immediately available thereafter, as has been shown.

In the foregoing, it was assumed that the engine was warmed up to permit running at idling speed. Provision must be made so that the ability of the transmission to reach and maintain a zero drive ratio at idling is adaptable to considerable variation in engine idling speed, including not only the idling speed when the engine is hot, but also that when the engine is cold on a winter morning.

For economical operation, the duty of the automatic control is:

(1) to maintain a zero drive ratio when the engine speed is less than a certain maximum idling speed; and

(2) to raise the drive ratio when the engine speed is increased above the idling range, varying it so that the engine speed is kept substantially constant at a certain slow speed somewhat above the idling range.

Obviously this control function can be performed by any suitable constant speed engine governor. Ordinarily a governor maintains the speed of an engine constant by controlling the input; in this case the engine speed is to be governed by controlling the output. Tendency of the engine speed to increase, must raise the drive ratio, and vice versa.

Once the drive ratio has been raised above zero by the

automatic control, car acceleration can be obtained in only two ways, namely: by opening the throttle, and by reducing the drive ratio.

The duty of the driver-operable accelerator, therefore, is:
(1) to open the throttle, and then
(2) to reduce the drive ratio.

From Fig. 10 it is clear that the function of the accelerator relative to the drive ratio is to shift the point of operation from the curve marked 600 rev./min. toward the curve marked 3500 rev./min. as a limit. The function is the same whether temporary acceleration or top speed is demanded. The accelerator pedal of the new standard control system can be connected to the carburetor and to the transmission, or preferably to the governor control of the transmission, so that it will open the throttle and reduce the drive ratio, in approximately the right sequence and relative degree, for the desired results under any conditions.

Conclusion

Given a practical and economically possible continuously variable transmission, certainly there is nothing in either the automatic or driver-operable control for it that is beyond the reach of intelligent engineering. When brains enough have been used in the solution of the transmission problem, the transmission itself will not need a brain to permit any driver to get maximum economy, yet with maximum activity always available on demand, merely by "stepping on it". In other words, *both* optimum economy and maximum accelerative ability can be provided, together with a rational system of simplified control.

The improvement in all-round performance which can be reached is shown by the comparative summary of the foregoing analysis in Table 5, using today's Car 33 with the gear box as the basis for comparison.

The improvement due to streamlining is nil at low car speeds, but increases rapidly as the speed increases. The improvement due to the continuously variable transmission is greatest at low car speeds, and extends in large measure well up the speed range, and is still effective to a small extent in increasing the top speed of the car. While a continuously variable transmission alone can improve acceleration and economy more than 40 per cent each, its more important contribution is to permit the reduction in engine size by which economy can be improved still more.

As cruising and top speeds are raised by streamlining, a better solution of the transmission problem becomes more and more imperative, to avoid excessive engine speeds. Demand will exist for super-highway hounds—cars capable of 120 mi./hr. with lower roofs, less frontal area, less visibility for the scenery, powered with engines of 100 hp. or more, but it will be a limited demand from those wealthy or ignorant enough to pay the price. That price includes carrying around and feeding an engine twice as big and twice as hungry as that necessary to do 90 mi./hr.

"No one is interested in fuel economy", is an alibi, not a fact. Given two cars, equal in all other respects, one using twice as much oil and gasoline as the other, which would you choose? So would the public. Millions would be glad to buy a car that would average 40 mi./gal. with somewhat better accelerative ability, sufficient top speed, and a simpler and safer control system. Improved performance of this order is clearly within reach by combining a smaller engine and a clean job of streamlining with a continuously variable transmission properly controlled.

Rotating Wing Aircraft

(Continued from page 118)

3. The low-speed performance is superior to that of an airplane.
4. Airplane high speeds will probably be exceeded.
5. Control system is as simple and easy to use as that of the airplane.
6. First cost will be slightly higher, but maintenance and operating costs will be equivalent to that of the airplane.

Comparison of the criterion with that of the autogiro shows that the two are identical. In the author's opinion, the two machines have about equal merit, although the development work on the gyroplane has not been as detailed nor as extensive as that on the autogiro.

A table of comparative values will be useful in comparing the different aircraft discussed here, both with each other and with the airplane, and may be set up by assigning arbitrary weights for the airplane to the conditions stated in the criterion. Such a proceeding can be looked upon only as an expression of personal opinion, but should have value because it offers the most direct and simplest method of showing the relative position of the different machines and their possible and probable merit in comparison with the airplane.

The merits of the rotating-wing aircraft touched upon in this paper, as presented in Table 1, place the helicopter in an unfavorable position. The ability to rise and descend vertically is obtained by a great complication in the structure,

Table 1—Comparison of Rotating-Wing Aircraft

Item	Airplane	Helicopter	Cyclogiro	Autogiro	Gyroplane
Maneuverability and controllability	25	20	20	30	30
Reliability, emergency landing	25	20	30	30	30
Low-speed performance	10	20	20	20	20
High speed	15	5	15	20	20
Simple operation	10	5	5	10	10
Cost	15	5	10	10	10
Total	100	75	100	120	120

resulting in a serious penalty upon the high-speed performance. The problem of landing or taking-off at zero forward speed is only a little simpler than taking-off or landing at a speed from 20 to 30 m.p.h., and often the latter would be preferable, as in gusty weather. The existence of a wind of from 5 to 15 m.p.h. would destroy any advantage held by a helicopter over the autorotating-wing systems.

The merits of the cyclogiro, autogiro, and gyroplane disclosed by the criterion that has been applied to them place them in a very favorable light. The satisfactory fulfillment of the different conditions imposed in the criterion makes it appear that the era of extensive private flying may be greatly aided by the rotating wing. The removal of the hazards associated with the airplane's minimum speeds of from 50 to 75 m.p.h., combined with the decrease in size of the required flying field both for every-day use and for emergency, should stimulate the interest of the general public in these machines. The realization of the possibilities set forth in this paper should materially widen the field in which the aircraft manufacturer may expect to find users of his product.

Engine Design from a Wear and Maintenance Viewpoint

By J. B. Fisher
Chief Engineer, Waukesha Motor Co.

WHEN discussing wear that takes place in an internal-combustion engine, the reasons for wear must be divided into two broad classes. The first is wear due to certain features in design which the engine builder either incorporates or fails to incorporate in his engine. The second is wear that arises due to operating conditions imposed upon the engine by the owner, either willfully or through lack of proper instructions and general information on the operation of his car. These two broad classifications are not sharply defined; they overlap, and the responsibility for wear often rests on several parties. As usual in cases of joint responsibility, the buck is passed merrily about among the engine manufacturer, dealer, service station, owner, driver, and others. This mania for passing responsibility on to the other fellow is an outstanding characteristic of many men. They become so refined, so subtle in the art of buck passing that they fail to recognize the signals, nor can they distinguish clearly between forward and backward passing and whether the home team is advancing or retreating as the game progresses.

The designer feels that he has done his part if he has designed an engine to function within certain operating ranges, to meet average conditions in the hands of an operator whom he unconsciously credits with an expert knowledge of the functioning, the whims and the weaknesses of an internal-combustion engine. He is not at all concerned with instructing the ultimate owner of his engine regarding these details that are so close to him that he assumes they are facts of common knowledge. The manufacturer of automotive equipment feels that he has done handsomely by his customer if, somewhere—buried with a set of tools, a jack and an oil can—he has included an instruction book which all too often remains a sealed book to the buyer.

We would have little respect for a school system in which the pupils were given a set of books, a chemical or a physical laboratory, and told to dig out the course; but that is about as far as we get with imparting information to our customers. It would be quite an innovation for a car manufacturer to state in his bill of sale that the title would not pass to the buyer until he could pass a test to determine his familiarity with the things he should know about his engine. The general inference has been that the less the public knew about the construction and operation of its engines the less would it be inclined to monkey and tinker with them. I am beginning to question this, for I have long observed that the owners and operators who are best posted on the operation, and particularly the limitations, of their engines are the ones who are getting the longest life from their equipment.

The owner, finally, is inclined to accept the car as a finished product, requiring only occasional trips to the filling station for oil, gas and water, to keep it in perfect operating condition indefinitely. As he drives his shiny new car out of the dealer's show room, wear is the last thing in his mind. The salesmen certainly didn't say anything to him about it, and, with a gadget on the car to clean the oil and another to clean the air, how can there be any wear? Little Red Riding Hood, with her few simple questions to the wolf, was a hard-boiled prosecuting attorney compared with the trusting public and its naive acceptance of the wares offered to it, as long as they are wrapped in cellophane. With this brief statement in mind regarding the responsibility of the various parties concerned for such wear as may take place, we will discuss it briefly from the designer's viewpoint. We will allocate to him such responsibility for wear as he should justly assume; and, to the other parties involved, their share as it appears in the discussion.

If I were to pick the one item that causes the greatest wear on the greatest number of cars, I believe it would be the operating of engines at too low a temperature with too light a load factor. I realize that many engines are ruined by overspeeding, but I believe this is a small percentage of the total that require early overhauling due to low-temperature operation and too light a load factor. Cars that show excessive wear are all too frequently ones that operate almost entirely in city service, with a mileage of 40 to 50 miles per day. They are subject to frequent starts and stops and long periods of idling. In the winter months, the oil temperature drops to 50 to 70 deg. fahr., at which temperatures the dilution, encouraged by a low load-factor and the resultant poor compression and incomplete combustion, is not thrown off but remains in the oil to play havoc with every working part of the engine. We have observed even truck engines—operating

[This paper was presented at the Annual Meeting of the Society, Detroit, January, 1934.]

MANY engines are ruined by overspeeding, but Mr. Fisher believes that this is a small percentage of the total that require early overhauling due to low-temperature operation and too light a load factor. Cars that show excessive wear are frequently in city service, with a mileage of 40 to 50 miles per day. They are subject to frequent starts and stops and long periods of idling and, in winter, the oil temperature drops to 50 to 70 deg. fahr.

To eliminate or minimize wear the designer should provide manifolding that will operate over protracted periods without loading and give higher mixture-temperatures when idling, temperature control to minimum jacket temperatures of 170 deg. fahr., starting aids that will eliminate abnormal conditions caused by zero weather and means for keeping the oil hot then, corrosion-resistant materials for cylinders, pistons, rings, piston pins and crankshaft, together with axle and transmission ratios that will keep the engine loaded as heavily as possible and reduce operation under light loads to the minimum.

Other design factors include wear of valves, guides and seats; lubrication of exhaust-valve stems and guides; combustion-chamber shape; and connecting-rod and main bearing wear caused by speed, engine-structure deflection and dirt. The designer must decide on what type and size of oil cleaner and of air cleaner to use.

Mr. Fisher urges management to encourage designers to follow their designs through the patternshop, foundry and production problems, and intimately acquaint themselves with the causes and effects of wear and breakage.

in ash collection or similar extremely light-duty service in the cities in winter—in which only the use of auxiliary oil pans with an inch of insulation would raise the oil temperatures to a point at which the oil could maintain a normal viscosity. An engine that will give 80,000 to 120,000 miles in hard inter-city work before regrinding may need a regrind at 25,000 or 30,000 miles in short-haul city-service with frequent starts and considerable idling if these conditions are not anticipated and provision made to meet them.

Minimizing Cylinder Wear

The recent researches by the Dutch Shell Co. and English investigators point to the corrosive action of the by-products of combustion as a most important factor of cylinder wear. As the temperature of combustion falls rapidly on the power stroke, the layer of gas next to the cylinder wall chills below the dew point of the products of combustion and it condenses on the walls. These residual products of combustion contain gases of a highly corrosive nature. Under some conditions with certain fuels, thin films of rust form with incredible speed. Wear may occur here at six or eight times the rate observed under more favorable conditions. What can the designer do here to eliminate or minimize the wear due to unfavorable operating conditions?

It would appear that the following items should be given consideration as a means to obtain the desired improvements:

(1) Manifolding that will operate over protracted periods without loading and even give higher mixture-temperatures when idling for a few minutes for the frequent stops in city traffic.

(2) Temperature Control on the cooling system to assure minimum jacket temperatures of 170 deg. fahr.

(3) Starting aids that will eliminate the abnormal conditions encountered in the morning start in zero weather.

Many designers calmly shut their eyes to this feature, trusting to a battery in A-1 condition, high-test gasoline, a low-viscosity winter-oil and a skilled operator, to secure a start. Unfortunately, this combination is quite uncommon, which accounts for the intense activity of the tow car between 7:00 and 10:00 a. m. on sub-zero days.

(4) Means for keeping the oil hot in zero weather; either heat applied to the oil by the cooling water or possibly sub-pans for extreme conditions.

(5) Materials specified for cylinders, pistons, rings, crankshaft and piston pins, that will be more resistant to corrosion than merely average materials.

Ring and cylinder materials have been particularly neglected in this respect, a Brinell specification on the cylinder frequently being the only specification sought. Iron alloys are available which show a much greater resistance to corrosion than do ordinary cylinder liners, but some of these alloys are more adaptable to use a dry or wet sleeve construction than to casting in the cylinder block itself. Very thin dry liners are already in use in some foreign cars as original equipment, and thousands are installed in service. They represent a cheap and easy replacement proposition where wear necessitates new pistons, but high initial costs have prevented more widespread use.

(6) Axle and transmission ratios that will keep the engine loaded as heavily as possible and reduce operation under light loads to the minimum.

Automatic transmissions have already appeared in production abroad which result in more favorable load factors. It is to be regretted that the American driving public does not show a more receptive spirit and will not pay for four-speed transmissions. Here again it seems doubtful if we have succeeded in getting across to the car drivers the real reasons for and advantages of a four-speed transmission. Recent trends in tractor transmissions show a tendency to load the engine more heavily and reduce the engine speeds for certain operations requiring 15 to 20 per cent of the engine power developed at normal governed speed.

Vacuum Gages

One very successful bus operator installs a vacuum gage on the dash. The dial has that portion of the face from 3 lb. to

15 lb. colored red, and his drivers have been carefully coached to operate suitable ratios to keep out of the red at every opportunity, with highly satisfactory results in fuel economy and maintenance costs. I am not suggesting a similar device for motor cars but am citing it as a case for more favorable engine loading, as a plea for a wider use of engine speed—not the carburetor throttle—to meet the reduced power requirements at low and average car speeds.

Other Factors of Designing

Wear of valves, guides and seats, is a responsibility that is mainly up to the designer. While certain fuels and lean mixtures often cause wear, they are not the causes in the vast majority of cases. Seat wear can be combated in two ways, by removable seats or by the quality of iron in the block itself. The fact that one engine has removable seats and another model has none does not necessarily mean that the second engine is inferior to the first. The first may have very inferior metal in the block itself, with resultant cylinder wear, while the second may show an excellent record for both valve-seat and cylinder-bore life. Location of the valves, with relation to one another and the cylinder barrel, is possibly one of the designer's most important decisions, yet at almost every automobile show there can be found models in which valves are spaced so close to the cylinders that deformation of valves and bores adjacent to the ports is bound to show up quickly. Wear that results from such conditions, or from improper spacing and anchorage of cylinder-head studs, is certainly a responsibility of the designer.

Lubrication of exhaust-valve stems and guides is often left to chance, with the chances in favor of excessive wear. Engines stopped suddenly under full load at top speed must show oil well up toward the tops of the guides if wear is to be avoided. Exhaust gases are trying to find the shortest path to regions of lower pressure, and it is a great deal nearer past the exhaust-valve stems than through the long and tortuous path out of the exhaust manifold, exhaust pipe and muffler. The rigidity of the supporting bosses that hold the valve guides is seldom given sufficient attention; they are often left to distort as they will, instead of attempting to hold them concentric with the valve seat.

Exhaust valves are very frequently far too large, often being the same size as the intake valves. They can profitably be made 60 per cent of the diameter of the intake valves, with no loss in power and a vast improvement in exhaust-valve temperature. The exhaust-valve temperature is the controlling factor in maximum permissible compression ratios, and, though the ill effects of a hot exhaust-valve may be mitigated by aluminum heads and other means, still higher powers and economies are available, even with aluminum heads, if the designer drives hard toward cooling the valve to the utmost. The highest possible compression ratio without danger of pinging should be the designer's principal goal. The high ratios give the best economies, and every ounce of fuel that we can convert into power spells not only better economy but means just so many less B.t.u.s diverted from expending themselves in useless heat and wear. It should give a moderate rate of pressure rise to avoid undue shock under full throttle not only to the engine parts but to the power-transmission assembly. It should have as short a flame travel as possible to discourage the building up of temperatures that will induce knocking.

I saw a concise and pithy definition of knock recently: "Engine knock is an undesirable form of compression ignition intervening and interfering with the normal operation of the Otto cycle." With that definition in mind, one can appreciate the potential damage and wear that may result when knocking starts.

The chamber also should have the spark plug located in a favorable location for good idling to permit as economical idling and light-load mixtures as possible, for 90 per cent of the fuel burned in the average automobile engine is burned under idling and part-throttle conditions. In the intensive development for the maximum output per cubic inch, this latter fact is often overlooked.

There are two fuel-consumption figures with which the designer should be familiar on both his own and competitive engines. One is the relation of idling consumption to fuel consumption at the load of best economy. The second is the relation of the leanest mixture on which his engine will operate smoothly without missing in any cylinder at wide-open throttle and the setting required for maximum power. These two figures tell him in a broad way how well he has done his job of manifolding, combustion-chamber design and spark-plug location.

Wear of connecting-rod and main bearings, assuming an adequate supply of oil, is due primarily to three causes; speed, deflection and dirt. If his engine is going into an installation where the service, axle ratios and sales claims are going to entail excessive speeds, the designer may well acknowledge the limitations of even the best of high-grade babbitt bearings, particularly as regards its loss of hardness at high temperatures. He can resort to lead-copper or cadmium-base metals that will stand more severe punishment at high temperatures. In the connecting-rod bearings particularly, the thin precision bearings with copper-lead linings have shown up to better advantage in critical installations than have babbitted rods. That the bore of the rod should remain absolutely round under all conditions, particularly when tightening the rod bolts, requires no comment, although we have seen rods on which the nuts had to be prick-punched on the rod line before finishing, and carefully tightened to the same identical slot in the castle nut to prevent distortion and wear. He has pussyfooted around and around this problem many times, making more or less effective passes at it with 23-cent air cleaners and oil cleaners that will clean dirty oil quite well but for entirely too short periods. Some cleaners, tested when new, will fill up and start to detour the oil through the by-pass in less than 5 hr. The abrupt step found near the center of rod or main journals, one side of the journal being larger than the other, is caused by dirt, the dirt usually passing out one side of the bearing.

Elimination of Dirt from Oil

The designer often overlooks the importance of arranging oil grooves or vents so that such dirt as may enter bearings can be flushed out of the bearing by the oil itself. Oil grooves are often inadequate, and, with inferior oil, small grooves and small holes feeding important bearings become closed. Designers think in terms of nice clean oil; not in terms of the diluted, carbon-and-dirt-laden mixture that furnishes the lubrication of the average engine.

The designer here must decide what type and size of oil cleaner is sufficient for his engine. No matter what size he chooses, he can rest assured that it will be too small. He should determine how long the cleaner will function without attention, preferably 10,000 miles at the least. If his engine must operate under conditions where oil temperatures rise to impossible figures, he cannot pass the buck along to the

oil company but must include in his design features that will keep the temperature within reason.

The selection of a proper air cleaner is a difficult problem on account of the wide range of conditions under which his product must operate. An air cleaner that may be fairly satisfactory in the eastern or northern states is often entirely inadequate in the South and West, and it is doubtful if he can meet these conditions with one type and size of air cleaner. The tractor industry is thoroughly conscious of the scrupulous attention that must be paid to the subject of air cleaners, but the passenger-car fraternity is too often inclined to form its idea of air-cleaner requirements from local atmosphere. The local atmosphere, by the way, is not as clean as may appear. During winter, the average monthly fall of soot in Detroit will vary from 600 to 900 tons per square mile. This soot, especially that from industrial stacks operating under forced draft, is hard, abrasive material, and air cleaners on compressors in power-house service often triple the cylinder life. Let the designer drive his product mile after mile through Idaho roads with the dust 8 to 10 in. deep, or through a California Santa Ana sand storm, which removes the finish from a car as would a first-class job of sand blasting, and his perspective on air cleaners undergoes a remarkable change.

Damage Caused by Dust

An experience that came to our notice in Oregon is enlightening in this connection. A bus engine had just been equipped with new pistons and cylinders and started on a trip up the Columbia River highway. It encountered an exceptionally severe dust storm, a storm so severe that is was noticed 500 miles out at sea. The bus made a round trip of 380 miles. When it returned to Seattle, a 0.035-in. feeler would fall through the clearance between the piston and cylinder.

Another case was that a fleet of ten trucks operating on a road-building project near Spokane. The engines were continually laid up due to wear. The owner, becoming exasperated and suspicious of his truck drivers, finally instructed his car driver to take the car and follow the trucks all day long to see if he could learn anything. He did. In ten days the car's operation deteriorated so that it was in worse condition than were the truck engines. The contractor drove it into the dealer's place for a check-up. Later, he was called to the dealer's shop and cross-questioned as to what he had been doing with his car. The damage done by trailing his trucks in clouds of dust was so appalling that he promptly stationed one man at the loading pit with 200 heavy woolen bags which just fitted over the air cleaner supplied as original equipment. This man's sole duty was to lift the hood on every truck as it drove up, remove the dirty bag and install a clean one. He finished his contract without further delay due to engine wear. All of this wear could have been eliminated in both cases by adequate air cleaners of the proper type for this territory, though not necessarily penalizing eastern cars with expensive air cleaners.

Wear of cylinders and pistons caused by dirt and dust can only be combated successfully by eliminating the cause. Very hard irons, hardened steel, and even nitralloy, offer little resistance to abrasives. In some recent tests of various materials in tractor cylinders, the wear of the hardest material was 70 per cent as much as that of medium-grade iron. Special alloys are of great value in resisting the corrosive action described previously in this paper, and the designer must keep these two principal causes of wear clearly segregated in his analysis if he is to combat them successfully.

Still another cause of cylinder wear, while recognized by engine builders for many years, has only recently been receiving active consideration from designers. This is wear caused by explosion pressure getting back of the top ring and forcing the ring out against the barrel. Wear due to this cause is particularly severe in engines which are subject to frequent and rapid applications of load, as in city bus-service where heavily loaded buses must be accelerated many times per hour. Explosion pressures applied to pistons moving at high speeds do not seem nearly so disastrous as the same pressures applied suddenly to pistons moving at low idling speeds. If you wish to show up wear rapidly in cylinders in a test, do not set the engine up and run it constantly at a high speed. Put it on a cycle that changes abruptly from a low idling speed to full load at moderate speed 20 or 30 times per hr., and the wear will be all that could be desired.

Some special rings have appeared which, in a large measure, neutralize this wear. The wafer rings, apparently because of the downward pressure on the upper face being far more than the relatively lower total pressure back of the ring, are effecting considerable reductions in cylinder wear. Due to their narrow width, they are less subject to rounding over on the face, which causes wide rings to lose their efficiency. The load effect of blow-by due to cylinder distortion has been graphically described in a recent paper by Ralph Teetor. I believe that anyone who has done much investigating with dummy heads in measuring barrel distortion after the cylinder-head nuts have been pulled down tightly will agree with his findings. A careful check-up of this nature should be included as standard procedure in any new model, provided that the engineering department can keep two jumps and a half ahead of the sales department.

Designers Should Be Fully Informed

If an engine designer is steadily to improve his technique in his work, he must be fully conscious of the foregoing points and many others of a similar nature. I do not mean that he must constantly be reminded of his mistakes and shortcomings until he loses confidence in himself and secretly doubts his ability to design a corn sheller. I believe it to be the duty of management to see that the designer is kept thoroughly posted on exactly what his product is supposed to do by first-hand observations, and to aid him in discovering and eliminating the reasons for wear or customer dissatisfaction. Too often the designer is in about the fourth or fifth-line trenches, when he should be in the first, to receive at first hand the reaction from the field. A great aid to such an end is a proper grouping of the departments with which he should be closely affiliated.

An ideal engineering department should have the pattern shop, casting inspection and returned-goods department, located so close to each other that they would be sitting in one another's laps.

Conscientious designers are keenly alert to measure the degree of success their product is achieving. We should strive, therefore, to see that they are actually watching their designs take shape in the pattern shop, noting the foundry and production problems involved, and seeing most clearly and intimately the causes and effects of wear and breakage. This is a final responsibility that rests on management, in acquainting and carefully schooling its designers with the thousand and one little details on which a successful design must rest.

Resilient Mountings as Applied to Automotive Engines

By Alex Taub
Development Engineer, Chevrolet Motor Co.

VIBRATION formerly was classed as such without much thought as to the determination of its sources, Mr. Taub states, and then came isolation of the various causes. The first two vibrations to be segregated and vigorously attacked were the secondary inertias of reciprocating units and torsional vibration. The development of the six-cylinder engine was among the earliest attempts to eliminate secondaries, and it was also the earliest producer of torsional vibration.

Dynamics, combustion roughness, torsional roughness and structural weakness, are a few of the contributing causes of engine roughness. Consideration must be given to all these factors if an engine is to be considered inherently smooth, and each is analyzed.

Engine mountings should have low resistance to rotation about the longitudinal principal axis and to rotation about the vertical axis through the center of gravity, together with minimum shift of affective principal axis and vertical axis. The kind of material used in engine mountings should be determined after considering the overall characteristics of the mounting and the durability of those characteristics.

THE purpose of this paper is to provide a means of looking backward over the trail of the development of resilient mountings. We expect to bring out, at least, for those interested, a few fundamental principles that we believe cover the business of proper mountings.

We will briefly touch their history, from mechanical to rubber, including rotations of the engine movement, frame rigidity and material specifications.

The observations included herein are personal and do not represent the opinion of any group; as such, they can be taken for what they are worth. The data are based upon a review of the work others have done in the industry with reference to the elimination of engine sensation, this review being aided by experimental work. We wish also to acknowledge the cooperation of various engineering organizations throughout the industry for data furnished to me for study.

Resilient mountings have been used in various industries for many years, the primary use being for insulation against noise and vibration. The earliest attempts in our industry were intended for insulation. Examination of correspondence from various engineers indicates how similar the history of all concerns really is.

Originally, the engine was set down in the frame with or without brackets and bolted down at convenient spots. Later, springs were added at the front to limit the crankcase distortion as well as to minimize vibration. In some cases three-point mountings were used; however, other organizations frowned on this practice because it "sacrificed" frame rigidity. Sub-frames came in for consideration. In one particular case, the Studebaker Corp. produced a car with a flexibly mounted sub-frame which rocked fairly freely.

Fabric liners were introduced between mounting brackets and frame and these liners were gradually changed to ever-increasing flexibility by adding to and/or softening the material used. To further increase the local flexibility, rubber in tension was introduced.

Engine movement or "flop" on the road became a factor, and the need for designing frames to stand on their own rigidity became apparent; then, an accomplished fact.

With the frame stiffened and the front-end weaving covered by special designs that eliminated radiator, head-lamp and fender movements, engines could be and were mounted with increased flexibility. An entirely different investigation was introduced and merged into the mounting picture, which merger brought about consideration of the *position of the mountings*. This other investigation had to do with the business of vibration and its *source*. In the early days, vibration was classed as such without much thought as to the determination of its sources. Then came isolation of the various causes.

The first two vibrations to be segregated and vigorously attacked were (*a*), secondary inertias of reciprocating units, and (*b*), torsional vibration.

[This paper was presented at the International Automotive Engineering Congress, Chicago, September, 1933.]

Elimination of Secondaries

The development of the six-cylinder engine was among the earliest attempts to eliminate secondaries, and also the earliest producer of torsional vibration. More than twenty years ago Lanchester developed mechanical antidotes for these vibrations; namely, the "bob-weight" counterbalance for secondaries and the torsional damper for long six-cylinder crankshafts. The "bob-weight" counterbalance was used in this country, but not extensively. The cure was considered by some to be worse than the disease. Dampers are in general use today. The battle of the secondaries, however, went on for many years.

The development of the 90-deg.-crankshaft V-Eight, as used in the Cadillac for years, was the result of that company's efforts to eliminate the vibration caused by secondaries from their original 180-deg.-crankshaft V-Eight. The battle against this form of vibration was not confined to our industry. An outstanding and interesting example is to be found in the electric-refrigeration industry.

The possibility of elimination of secondaries by mounting began to interest engineers. Three outstanding examples appeared. The electric-refrigeration industry produced a complete commercial result. The compressor and its driving unit were set on opposite ends of a frame, which frame was very flexibly supported at the center of gravity of the whole unit on a base. Rocking was limited by flexible bumpers. This unit operated with commercial freedom from secondaries; however, the unit rocked violently as a whole about its mountings when starting.

Engineers became imbued with the idea that the 90-deg. crank for V-Eights, with its attendant heavy and costly counterweights, could be eliminated, as well as its now corrected inherent distribution weakness. Engines with 180-deg. cranks were mounted to permit movement about the polar axis of the engine. These mountings were fairly flexible, but by no means excessively so. In conjunction with the mountings was a mechanical device designed to eliminate the effect of transverse movement of the powerplant. This unit was designed to provide zero change in load on the frame by means of a cam-operated plunger moving at twice engine speed, and was attached to the frame. Direction of movement was phased to offset engine movement. This device was originally developed for four-cylinder engines by C. E. Summers and R. K. Lee, at that time of the General Motors Research Corp., and was adapted by Summers to the Oakland Eight which was quite as smooth as the 90-deg.-crankshaft V-Eights.

Another interesting example was a four-cylinder-engine solution of secondaries, this being the next to appear. Here again the unit was mounted to swing or move about or around the polar axis. In this case the mountings were extremely flexible, the engine moving considerably, making necessary special consideration for connections of various pipes and leads to the engine. We must admire the courage of the organization which commercialized extreme flexibility and produced a fair commercial result in the business of smoothing out the four-cylinder engine.

Three examples of the commercial elimination of secondaries have been cited, but they do not exist today. The electric refrigerator uses a new type of compressor that is inherently smooth. The Oakland Co. uses a Line-Eight engine that is inherently smooth. The four-cylinder proponent uses a six-cylinder engine that is inherently smooth. Each, however, has applied his mounting knowledge to his inherently smooth unit with considerable success. Merchandizing expediency is quite often more potent than engineering ingenuity.

Engine Roughness

A few of the contributing causes of engine roughness are:

(1) Dynamics; including running balance or out-of-balance
(2) Combustion Roughness; including rate of pressure rise and high brake mean effective pressure
(3) Torsional Roughness
(4) Structural Weakness

Consideration must be given to all of the foregoing factors if an engine is to be considered inherently smooth. At best we can only expect engine mountings to add that portion of "silk" thought necessary in modern motor-car operation. To overburden the mountings with an inherently rough engine will mean excessive softness for the mountings and a low durability.

Engine roughness during car operation can be divided into two ranges; (*a*) below 20 m.p.h., or torque reaction, and (*b*) above 20 m.p.h., or range roughness. These conditions or elements may be treated structurally to a certain extent. Torque reaction, being a function of the engine impulses per car mile, can be washed out of the picture by increasing the number of cylinders. Range roughness, though more complex, does react to treatment along the lines of rigidity, correct balance and low reciprocating values. However, in spite of all we may do, shock-absorbing mountings are necessary.

Two apparent facts stand out as a result of careful observation. They are:

(1) Low resistance to rotation about the principal axis provides low torque-reaction frequencies.
(2) High lateral freedom—transverse of the crankshaft axis—provides the maximum of smoothness throughout the speed range.

The first can be readily supported mathematically. The following data were developed by William Samuels, of the Chevrolet engineering department.

"A six-cylinder engine may be assumed to be a three-cylinder engine having all cylinders in one central plane, and having in each cylinder reciprocating weights of twice actual value, with cranks spaced at 120 deg.

"Superimposed upon each other a regular torque wave is produced which for all practical purposes may be considered a regular sine curve with three full waves per engine revolution. The resulting frequency is therefore three times the number of revolutions.

"The frequency of engine movement can be determined as follows, reference being made to Fig. 1.

"The weight of the rocking body (engine and transmission assembly, Confederate, 1932) is 660 lb. The term 'body' refers to the rocking body, that is, engine and transmission assembly. The moment of inertia of the body is

$$I_1 = 25{,}087 \text{ lb.-in.}^2$$

$$I = \frac{I_1}{g} = \frac{25{,}087 \text{ lb.-in.}^2}{386 \text{ in. per sec.}^2} = 65 \text{ lb.-in. sec.}^2$$

"Radius $R = 11.25$ in. and $R^2 = 126.6$ in.^{2}

"The rate of each spring is 50 lb. per 1/16 in., or 800 lb. per in.

"The total spring constant, c, equals 2×800, or 1600 lb. per in.

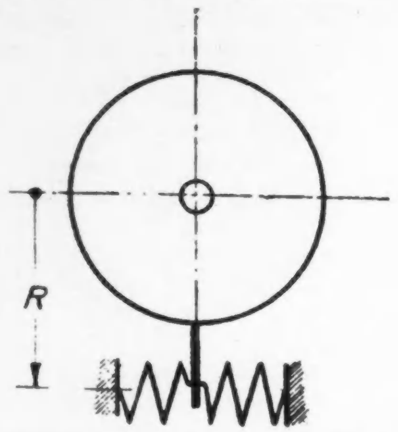


Fig. 1—Frequency of the Engine Due to the Support

"The mass of the body, reduced to radius R, is $m = I/R^2$. Substituting the foregoing values gives $m = 65/126.6 = 0.5135$ lb.-in. sec.^{2}

"The inherent speed of the system, V, is $\sqrt{(c/m)} = \sqrt{(1600/0.5135)} = \sqrt{3118} = 55.8$ radians per sec.

"The frequency, $f, = V/2\pi = 55.8/2\ (3.1416) = 8.88$ per sec.

"The frequency, $f_1, = 60\,f = 60 \times 8.88 = 533$ per min.

"The engine will vibrate, due to this suspension, at an engine speed $n = 533$ r.p.m.

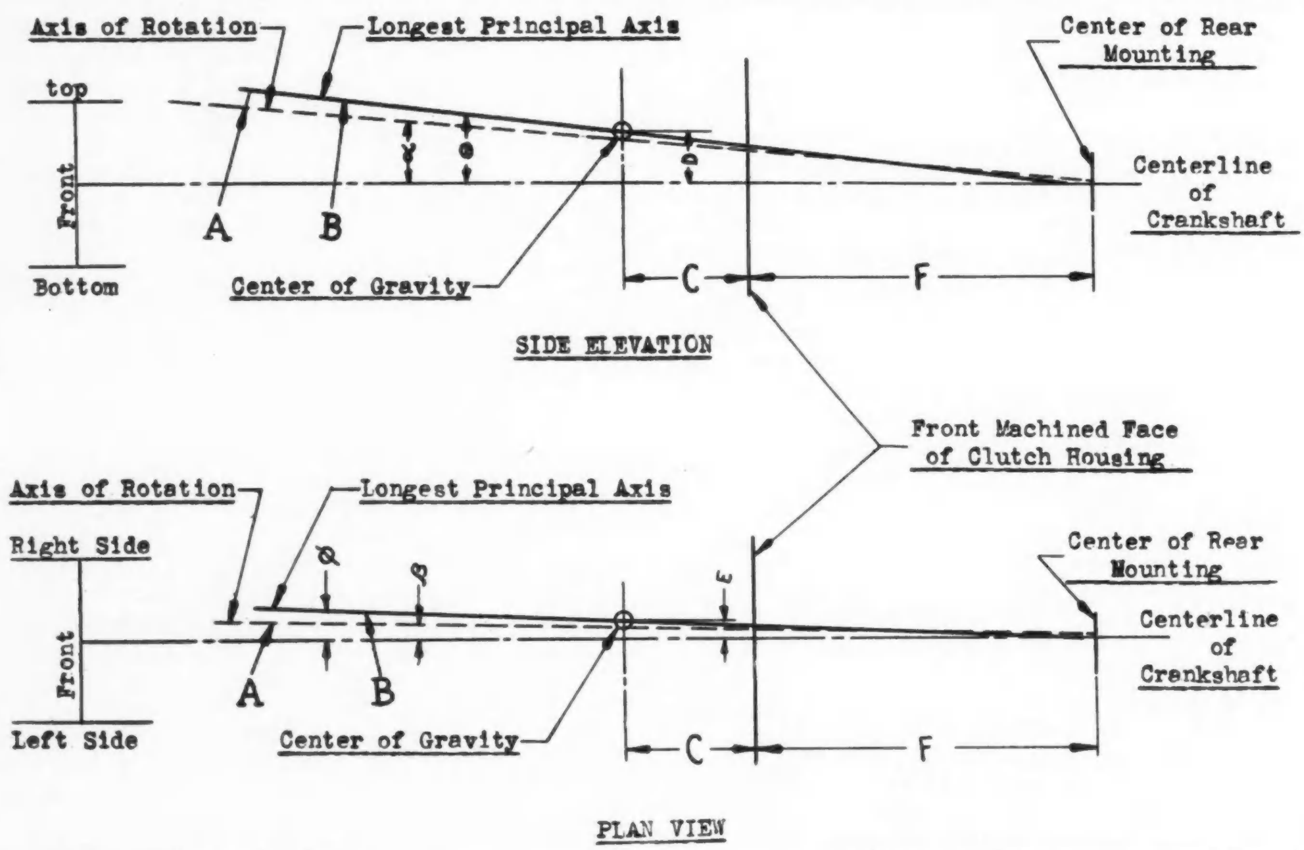
"By transformation, frequency f_1, is $f_1 = \sqrt{c}\,R\,(9.55/\sqrt{I})$ per min.

"For the values $I = 65$ lb.-in. sec.^{2} and $R = 11.25$ in., $f_1, = 13.33\sqrt{c}$ per min. From this equation a chart for varying spring constants, c, can be drawn. For a radius r, instead of R, the ordinates should be multiplied by r/R, or $r/11.25$ in this instance."

Polar Axis Discussed

Perhaps at this point it would be well to discuss the principal axis, or polar axis, of an engine. There has been much discussion of the location of this axis and the factors controlling its position. Obviously, the position is greatly affected by the resistance and location of the engine mountings.

Broadly speaking, on a unit powerplant, the longitudinal principal axis will run through the center of the U-joint and intersect the center of gravity. This is not exact but is much closer than the actual axes of rotation will be when the engine is rotating on its mountings, no matter how accurately placed. We have carefully checked the axes of rotation of many engines in cars and rechecked the engines free of their



Engine	α		β	Θ		φ		C	D	E	F
	Deg.	Min.	Min.	Deg.	Min.	Deg.	Min.				
1	7	45	0	9	1	1	4	8 5/32	4 21/64	1/8	18 21/64
2	9	15	44	10	51	1	43	6 19/64	3 21/32	43/64	17 3/16
3	6	25	59	8	38		51	8 21/32	3 57/64	17/32	17 7/32
4	6	4	22½	7	2½	1	30	9 9/16	4 1/32	23/32	20 7/16

Fig. 2—Variation of the Position of the True Principal-Axis Location

Location of longest principal axis, axis of rotation, and center of gravity of engine and transmission assemblies.

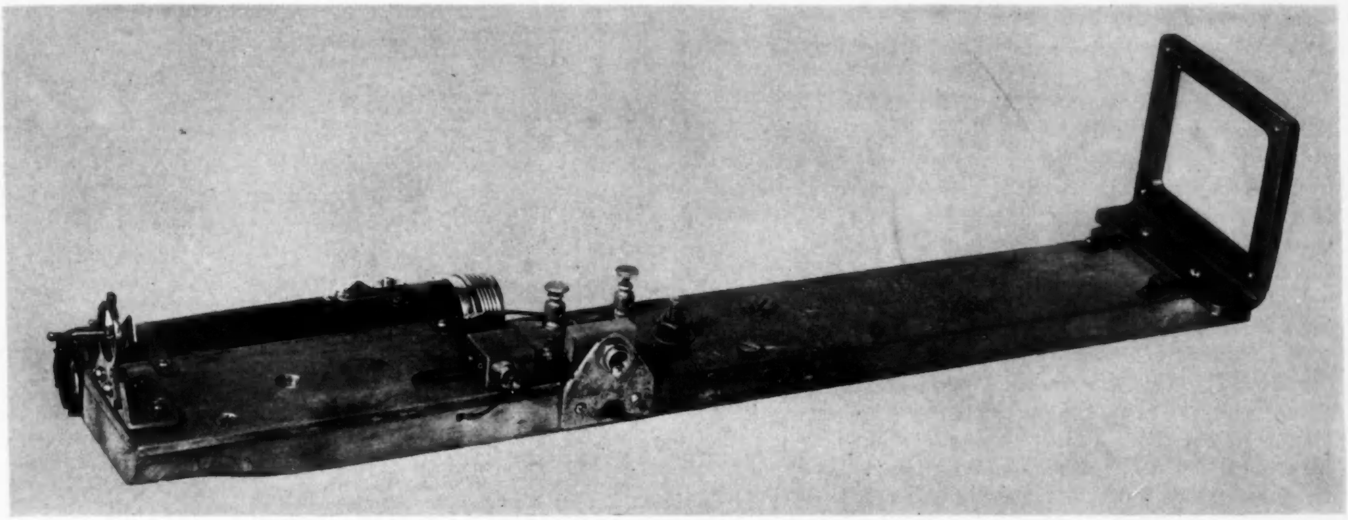


Fig. 3—Optical Indicator Affording a Simple Means of Recording Photographically Movements of Mechanical Units Under Operation

mountings. In no case do they coincide. Fortunately, however, extreme accuracy of mounting position is not essential. Reasonable accuracy of position is necessary.

Fig. 2 gives some idea of the variation of position of the true principal-axis location from our previous assumption. The intersection of the U-joint center and the center of gravity is shown as *A*, and the true principal axis *B* is obtained by hanging the engine from its end and changing the location of the point of suspension to obtain the minimum moment of inertia. Four entirely different types of engines are shown.

Resistance to rotation about the principal axis, of which there are more than one, is the controlling element for engine-sensation absorption. Low resistance to rotation will result in operating smoothness. This of course may mean low internal rigidity for the mountings. Durability of the mountings depends to a great extent upon this internal rigidity. When too flexible, they may not stand up under engine movement due to road shock; therefore, we cannot blindly take advantage of the fact that low resistance to rotations is beneficial, for instance, in reducing the frequency of torque reaction.

Following are some figures on resistance to rotation and the resulting torque reaction of various 1932 cars.

A	*B*	*C*
1500 ft.-lb. per deg. 15 m.p.h.	2500—23 m.p.h.	5000—35 m.p.h.

A, *B* and *C* represent three ranges of resistance to rotation during 1932. The *A* group is the average of a fairly smooth group. Group *B* is the average that was barely commercially smooth. Group *C* is the average that we believe to be non-commercial having a torque reaction from 5 to 35 m.p.h. The data are given here to record these values for the 1932 average for cars with average or less flexibility.

During the early part of 1932 an investigation was made on a series of cars with extremely flexible mountings to determine the relative flexibility of these jobs as compared to the then conventional flexibility. Assuming that a flexibility of 1500 ft.-lb. per deg. was 100 per cent, the four-cylinder super-flexible job's rotational resistance was down 2145 per cent.

Comparing a six and an eight-cylinder engine with mountings similar to the foregoing we found, assuming a 1932 average of 1500 ft.-lb. per deg. for a six-cylinder engine as 100 per cent, that the four-cylinder is 2145 per cent down, the six-cylinder is 727 per cent down, and the eight-cylinder is 233 per cent down.

Since these three cars were satisfactory jobs from the smoothness standpoint, the indications are that a 3:1 ratio exists for comparative results between a four and a six, and also between a six and an eight-cylinder engine. However, in 1932, it is well to note that this particular eight-cylinder engine was mounted with two and one-half times the flexibility of the average six-cylinder engine.

Degree of flexibility or resistance to rotation is important, regardless of how it is obtained. However, the more accurate the location of the mountings is, the lower the resistance to rotation can be for the maximum of firmness in the mounting brackets. The resistance to rotation picture of today has changed. It is down considerably from 1932. In some cases without sacrifice of mounting firmness; in others, by application of very soft pads. Obviously, the softer the detail mountings are, the less is the need for definite location. However, we believe durability will force consideration both for decent location and limited flexibility.

Checking of the rotational resistance of several 1933 cars indicated the following: Overall rotation resistance—average six-cylinder—1100 ft.-lb. per deg. Torque reaction, 5-7 m.p.h.

Flexibility	Ft.-Lb. per Deg.
Maximum, six-cylinder,	500
Average, eight-cylinder	1100
Maximum, eight-cylinder,	725

The torque reaction in the eights appeared eliminated, although, in the maximum-flexibility jobs, mountings varied considerably in construction. Durability in one case has been sacrificed materially where low resistance has been accomplished without regard to position. The mountings are forced to absorb the effects of shifting the effective principal axis, and are too soft to stand up under road shock.

If the mountings due to position and resistance disturb the mass balance, then the principal axis shifts. The amount of principal-axis shift may be assumed to represent the extent of position error in the mountings.

Range Smoothness

Range smoothness is affected by many things; some may be offset by insulation, others by carefully designed resist-

ance, and still others must be eliminated at the source. We will deal with that portion of range roughness that can be affected by mountings.

Three years ago, during a study of crankcase deflection on the dynamometer under fairly high brake mean effective pressure, we found by actual timing and measurement at high engine speeds that the plan view or transverse deflection of the case was in the form of an "O.G." curve at the front bearing. This particular engine had a "bump" usually associated with high brake mean effective pressure. A metal mounting was installed at the front end that provided transverse freedom but restricted the up-and-down movement. This mounting eliminated the sharp O.G. in the case and the "bump". This same bracket was installed in a 1931 car which had metal mountings, with very gratifying results. Transverse freedom was determined to be vital for range smoothness. This is a fact that has been brought home to us in many ways since, and we will endeavor to support this by other data.

The General Motors Research Laboratory has developed a simple means of photographically recording movements of mechanical units under operation. Fig. 3 is an optical indicator which consists of a concave mirror mounted so as to have relatively unrestrained tilt in two directions. This corresponds to horizontal and vertical movement. A shutter mechanism electrically controlled was placed between the source of light and the mirror, which reflected a spot of light on a screen or film as desired. Two crankshaft revo-

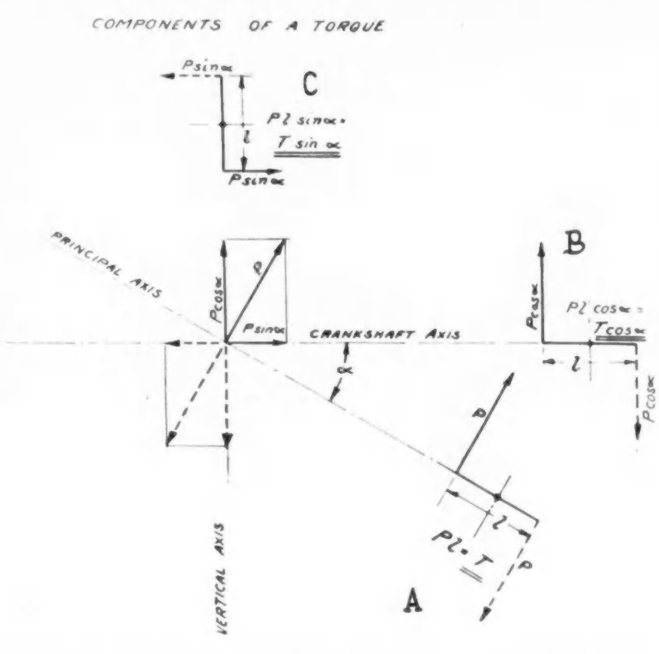


Fig. 5—Illustrations Showing That the Angle between the Polar Axis and the Source of Vibration Causes a Resultant Couple about the Vertical Axis Through the Center of Gravity

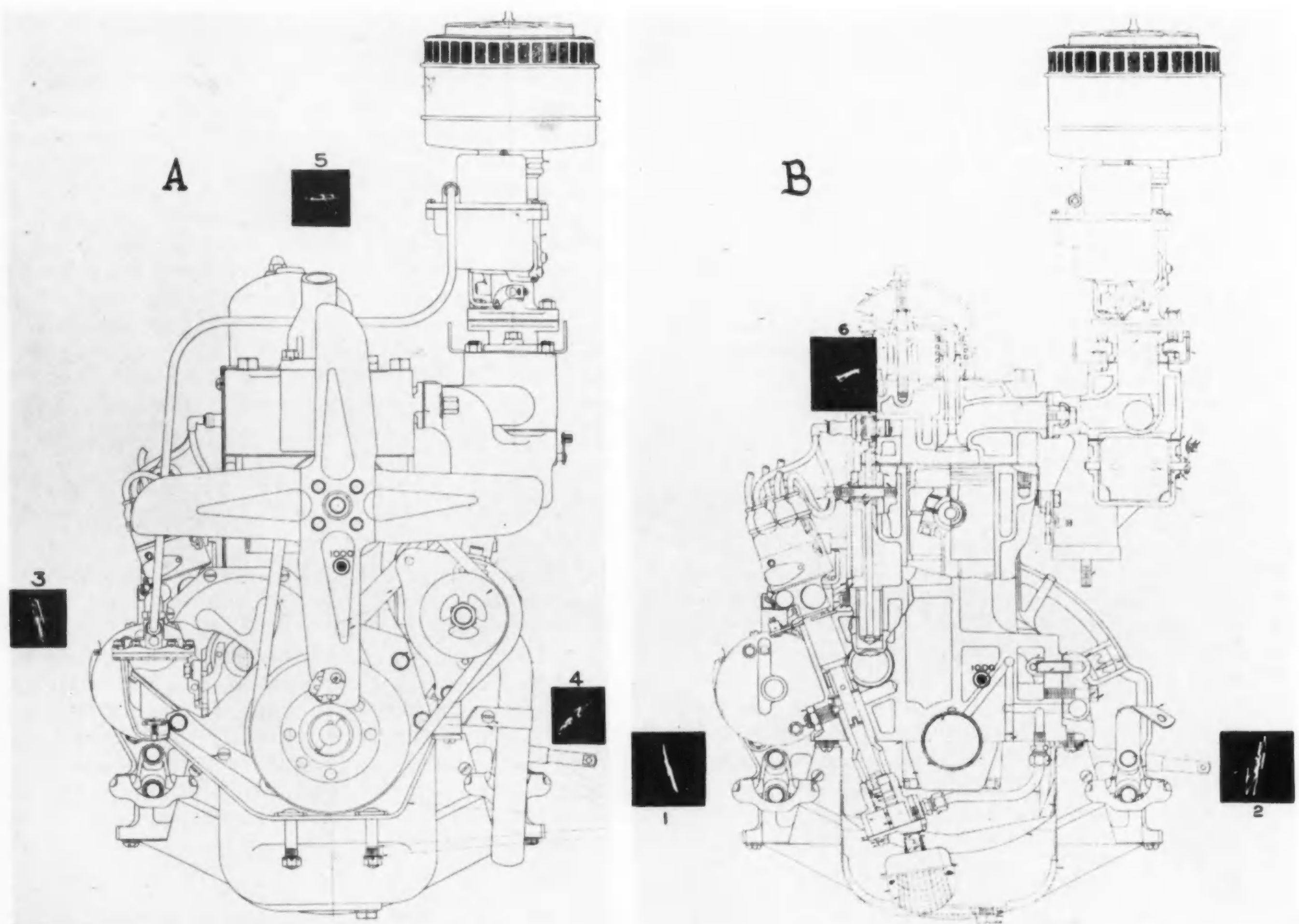


Fig. 4—Diagrams Indicating Actual Paths of Light Beams Moving with the Engine

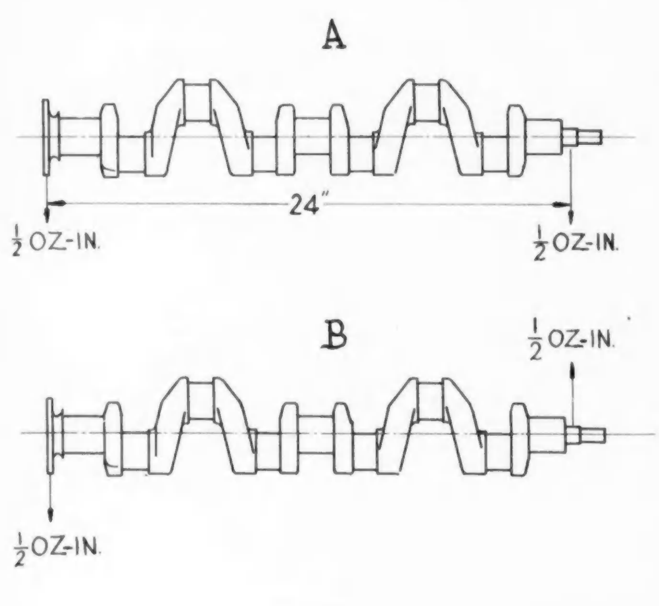


Fig. 6—Diagrams Showing How Great Dynamic Out-of-Balance May Become

lutions for each shot was allowed. A stroboscopic effect can also be arranged to slow down the light beam.

In Fig. 4, views *A* and *B* indicate actual paths of light beams moving with the engine. Note their relative direction with reference to the polar-axis center. There is not the slightest doubt that the engine rotation is about the principal axis. This axis is an oblique line and the excitation of the movement is from the crankshaft. The angle between the polar axis and the source of vibration causes a resultant couple about the vertical axis through the center of gravity as shown in Fig. 5, which shows also the arrangement by Samuels of couples that establishes the existence of a rotating force *C* about the vertical axis generated by the oblique rocking action as at *A*. This couple causes the engine to oscillate about the vertical axis. Investigation has proved that this movement about the vertical axis is equal in importance to rotational movement about the horizontal axis, and, if not provided for either by design or accident, range smoothness cannot be had.

Dynamic out-of-balance is prevalent in most engines today to some extent, and will not be eliminated until assembly balance or selected balance is in more general use. Dynamic out-of-balance is a direct force tending to turn the engine about its vertical axis. This unwelcome force adds its might to the couple produced by the angularity of the principal axis and makes essential the provision for a limited transverse rotational movement.

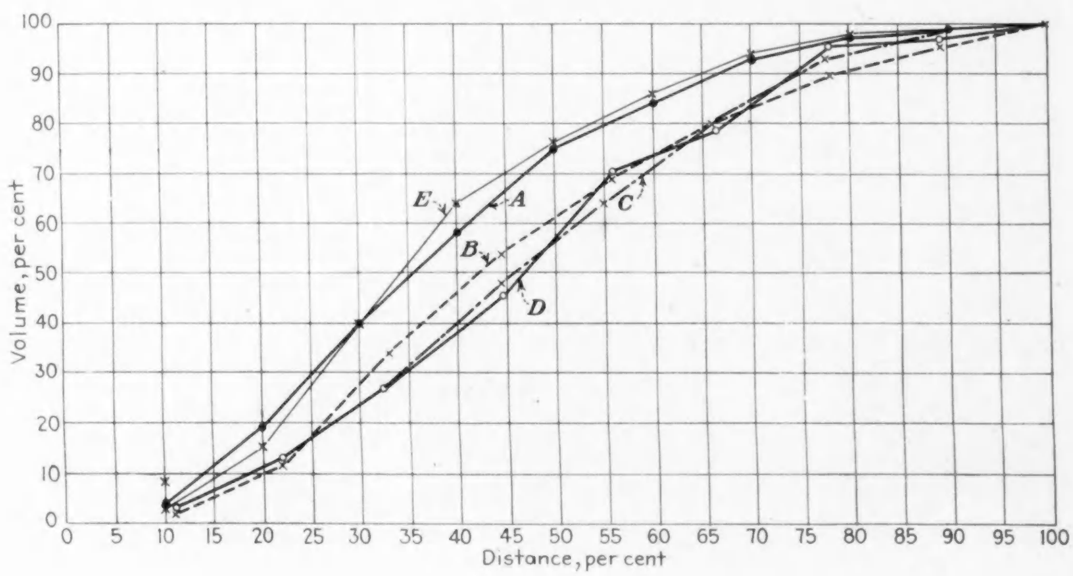
It is extremely difficult to eliminate dynamic out-of-balance at the source, and an out-of-balance of 4 oz.-in. will penetrate the best of mountings. Consider for a moment a well-balanced crankshaft held down to ½ oz.-in. on each end. If this out-of-balance is as indicated in Fig. 6-*A*, wherein the out-of-balance in each end is in a plane, the out-of-balance will be the small total of 1 oz.-in. However, if the out-of-balance should occur in the form of a couple as in Fig. 6-*B*, the out-of-balance may total 12 oz.-in.[2] or more, depending upon the length of the shaft. Consider further the possible out-of-balance of the harmonizer, flywheel and clutch parts. The effect of each may be low, but the accumulation and position may combine to make a considerable effect.

Another source of trouble is crankpin-position tolerance, for radius and angularity. In the shaft-balancing operation, these small errors will be washed out in the correction of the balance; however, when rods are assembled to crankpins, the heavy end of the rod picks up the position error and this results in out-of-balance. The result may be static or dynamic and, added to the other possibilities, certainly gives us something to think about. Surely the effect on transverse rotational movement about the vertical axis is apparent, and so is the need for providing the offsetting flexibility. Assembly balance has been given consideration in several plants; also, selective balance, which is accomplished by assembling light and heavy sides to offset each other rather than permit the out-of-balance to accumulate.

Combustion-Chamber Roughness

Combustion roughness is an old offender against which considerable progress has been made. We must consider it here because it is an important factor in roughness. Engine mountings undoubtedly can overcome or absorb considerable

Fig. 7—Simple Comparative Curves That Give the Percentage Volume of the Combustion Chamber in Various Locations as Progress is Made Across the Chamber



harshness created by combustion; but, certainly, mountings cannot be expected to wash out all of the possible bumps that can be created by combustion.

The distribution of the combustion-chamber volume is still of extreme importance. Fig. 7 represents a series of simple comparative curves that, in my opinion, should be used by all engineers before calling a job done. These curves give the percentage volume of the chamber in various locations as we progress across the chamber; they were relatively simply obtained without calculation, and eliminate the necessity for developing the resultant time-pressure or combustion curve which depends for accuracy on a varying constant.

A simple cutting machine was rigged that could cut spherically. Segments of a plaster cast were progressively cut out, and the remaining volume checked each time. Curve *A* is from an L-head combustion-chamber and is ideal, being relatively smooth and indicating that the volume is distributed correctly, meaning that it is proportioned to give a smooth combustion-pressure rise throughout. This particular result has been repeatedly checked in cars and does give smooth operation. Curve *B* is from an overhead valve engine and is also ideal. There was a time when it was thought impossible to obtain combustion smoothness in an overhead-valve engine. It was assumed to be a prerogative of L-head engines. However, Curve *B* is proof enough.

Curves *C, D* and *E,* all show possible rough spots which could be eliminated with a little consideration. Curve *E* is from an inherently rough engine, although it has been materially helped by mountings. No doubt there may be other causes for the roughness of the engine represented by Curve *E.* However, the combustion curve certainly indicates need of adjustment of the combustion-chamber-volume distribution. Incidentally, this curve is from an L-head engine.

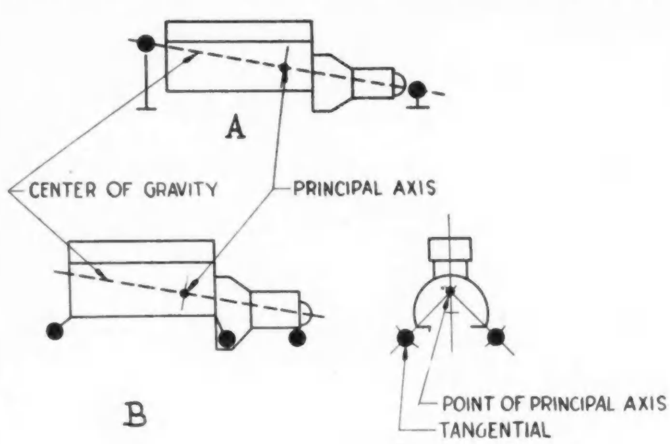


Fig. 9—Means for Approaching the Ideal for Minimum Shift of the Principal Axis

Mountings at Principal-Axis Centers Are Shown at *A*, and Mounting Tangential to the Principal Axis at *B*. These Two Groups Cover the Majority of Designs.

Fig. 8—Characteristic Light-Beam Chart of Engine Movement with High Compression

Fig. 8 is an interesting picture of a characteristic light-beam chart of engine movement with high compression. Note the wavy condition of the record. This condition did not exist with low compression. The lines were clear and smooth.

Torsional vibration is being adequately met by modern harmonizers with proper tuning. The types of harmonizers used vary, their desirability being a matter of opinion.

Structural weakness of crankshaft and of crankcase are broad problems, and require complete treatment in a paper on this subject. Resonance of either case or shaft will make itself felt and heard through almost any type of mounting. With this remark we will leave the subject until a more propitious occasion.

Engine Mountings

From the foregoing it is seen that what is desired for engine mountings is:

(1) Low resistance to rotation about the longitudinal principal axis.
(2) Low resistance to rotation about the vertical axis through the center of gravity.
(3) Minimum shift of affective principal axis and vertical axis.

There are several means of approaching the ideal for minimum shift of the principal axis. Fig. 9 shows *A*, mountings at the principal-axis centers and *B*, mounting tangential to the principal axis. These two groups cover the majority of designs.

Tangential mountings are being used fairly broadly, and we will confine our discussion and analysis to this type. They must either be capable of compound action, or must be located so that its one directional freedom will permit a rocking movement about the longitudinal axis, and also rotational movement about the vertical axis.

Tangential mountings at the front end are divided into two classes as in Fig. 10, two points as in *A*, or single point as in *B*. Undoubtedly, from the slow speed, the two or the single-point front can be equally effective. However, the effect during high speed must be considered, since it is under these conditions that rotation around the vertical axis is in force.

As stated, it is obvious that the detail construction can be compounded as in *C*, Fig. 10, or the position may be compromised as shown in *A* with somewhat increased softness. Further, we may use a support that is so soft as to be universal. With the latter, engine flop during rough-road operation may be the limiting factor.

The single-point mounting with its single direction satisfies both low-speed rocking and high-speed rotation. Being correctly placed, no compromise in softness need be made.

In *D*, Fig. 10, is shown an old type of mounting that incorporated a 5.5:1 ratio of horizontal movement to vertical, thus adequately restraining the engine against flop and providing relative freedom in a plane capable of satisfying the compound demand. There are many other designs that may do just as well.

Rear Engine-Mountings

Rear engine-mountings are more complex and single-point mountings for the rear have been lacking in durability although, like the front, they satisfy both rotations. The transmission support in *A*, Fig. 11, is a support for the rear, located under the transmission. However, for durability, we find that two mountings at the rear have been necessary.

The position of the rear mountings varies from ahead of the flywheel housing to the rear of the transmission. In the majority of cases they are located tangential to the longitudinal principal axis. However, this position does not satisfy the high-speed demand unless, like the front, they are com-

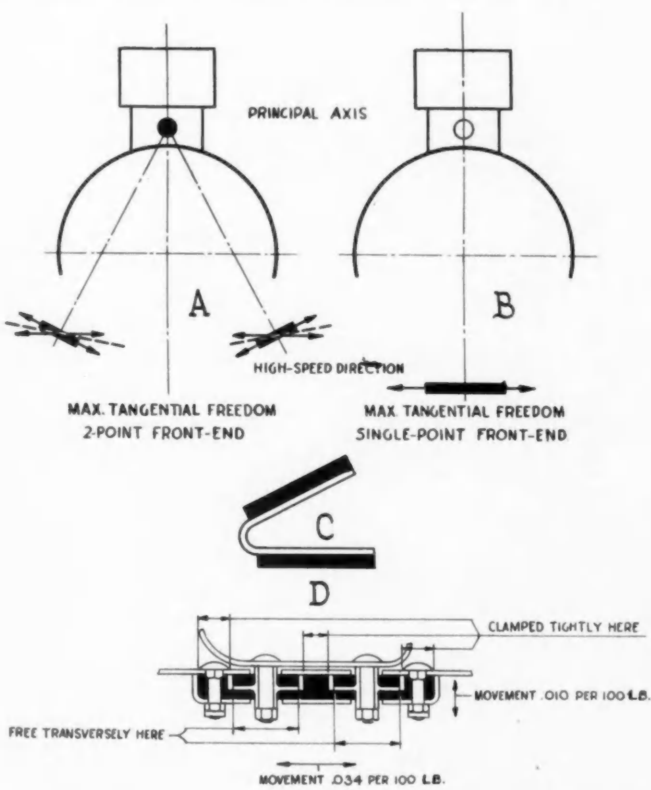


Fig. 10—Tangential Mountings at the Front End

A Two-Point Mounting Is Shown in Diagram *A*, and a Single-Point Mounting in Diagram *B*. The Detail Construction can be compounded as shown in Diagram *C*, or the position may be compromised as shown in Diagram *A* with somewhat increased softness. Diagram *B* represents an old type of mounting that incorporated a 5.5:1 ratio of horizontal movement to vertical

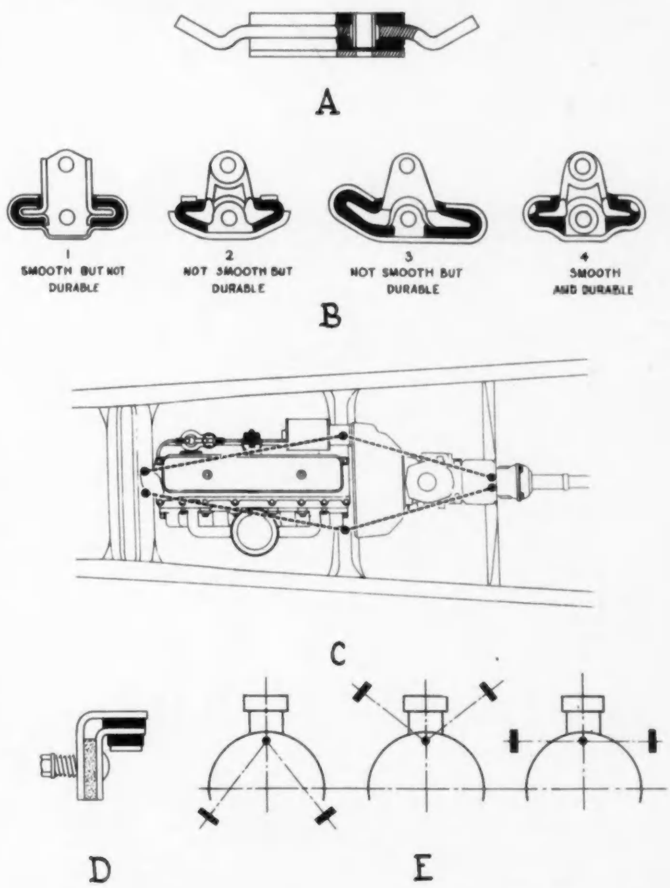


Fig. 11—Rear Engine-Mountings

A support for rear engine-mounting, located under the transmission, is shown in Diagram *A*, while Diagram *B* indicates a series of rear mounts made to obtain high-speed smoothness and located as shown in Diagram *C*

pounded or oversoft or in some other way made universal.

In *B*, Fig. 11, is indicated a series of rear mountings made to obtain high-speed smoothness. They were located as indicated in *C*. Only the designs permitting sideways movement were smooth. However, the slow-speed effect was not satisfactory. This series of experiments does serve to prove the importance of rotational movement about the vertical axis; in fact, in these experiments, we learned that it was more important than the rocking about the longitudinal axis.

To substantiate our thoughts further on the necessity for rotational freedom, the Inland Manufacturing Co. furnished us with a transmission support incorporating a large measure of friction as shown in *D*, Fig. 11. This definitely upset things by making the job rough through both the high and the low-speed range.

Looking at this whole problem we are forced to agree that we must provide relative movement about two axes which pass through the center of gravity. Then it is universal rotation at the center of gravity that is required, and the ideal would be a universal support right at the center of gravity. This is impossible, but a fair compromise can be made. The resistance to rotation should be balanced about the center of gravity. The relatively free plane of the mounts should be about the center of gravity, and the resistance should be in proportion to the distance from the center of gravity. In this way we not only provide the necessary direc-

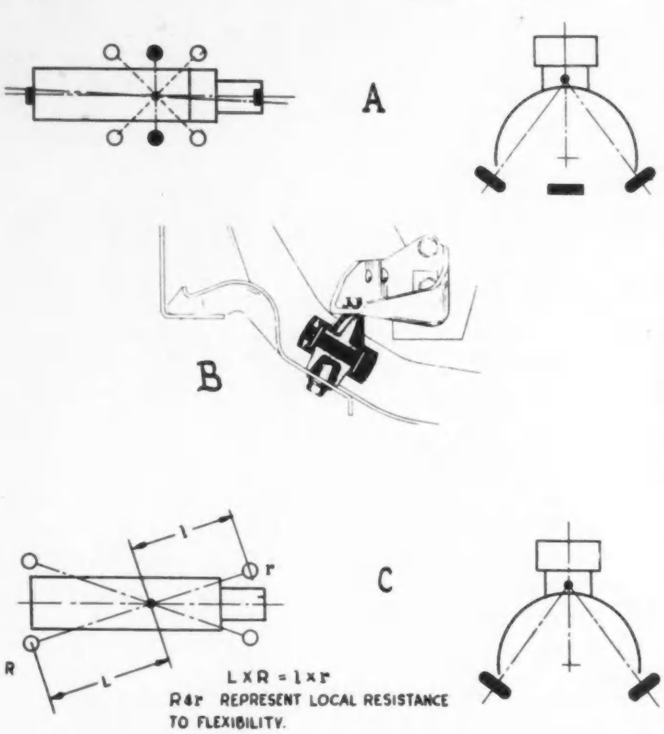


Fig. 12—Diagrams Showing How Balanced Resistance about the Center of Gravity Can Be Accomplished

Diagram *A* shows a single-point and direction mount as a steady rest front and rear, and universal mounts adjacent to the center of gravity tangentially placed about the principal axis, thus accomplishing relative freedom about both principal axis. Variations of the foregoing principle are indicated in Diagram *C*

tion desired, but we also cause the minimum shift of the vertical axis.

Fig. 12 includes several diagrams showing how balanced resistance about the center of gravity can be accomplished. In *A*, is shown a single point and direction mount as a steady rest front and rear, and universal mounts adjacent to the center of gravity tangentially placed about the principal axis. Thus, relative freedom is accomplished about both principal axes as shown in *B*. There can be many variations of this principle, as indicated in *C*. Much depends upon the chassis set-up and structure. For instance, consideration of the Hotchkiss drive and of the torque-tube drive requires separate treatment, since, with the latter, brake and drive reaction must be provided for.

Sample Mountings

A series of engine mountings is shown in Figs. 13 to 15. Fig. 13 represents the current Graham-Paige. Figs. 14 and 15 include a series of mountings made by the Inland Manufacturing Co., Dayton, Ohio.

Material

It is obvious today that insufficient knowledge of the material involved in engine mountings exists among the users of these parts. If fair control is to be exercised over the material that finds its way into our products, greater consideration must be given to the work that is going on in rubber laboratories, to establish worthwhile means of checking or inspecting material.

In considering the material used in engine mounting we find we must deal with two important items: (*a*) the overall characteristic of the mounting and (*b*), the durability of those characteristics. Dimensional checks of deflection under load, durometer or other hardness readings, will give an adequate check on (*a*), but (*b*) is not quite as simple since in it are involved not only the compound but every move in the process of producing the mount.

Until such time as the committees working on rubber specifications have definite recommendations to offer, we believe it is wise to make laboratory checks of several mountings under constant load and position, somewhat equal to operating conditions and at a temperature of about 150 deg. fahr. Under these conditions apply a stroking load at about 300 strokes per min. in a direction 90 deg. to the original static load. After each hour of stroking the rate of deflection of the sample should be checked. This will determine the comparative durability of the mounting characteristics.

Cooperation with the rubber source will usually bring improvement when desired. It is interesting to note that changes for durability improvement usually do not involve compound changes, but rather changes in processing.

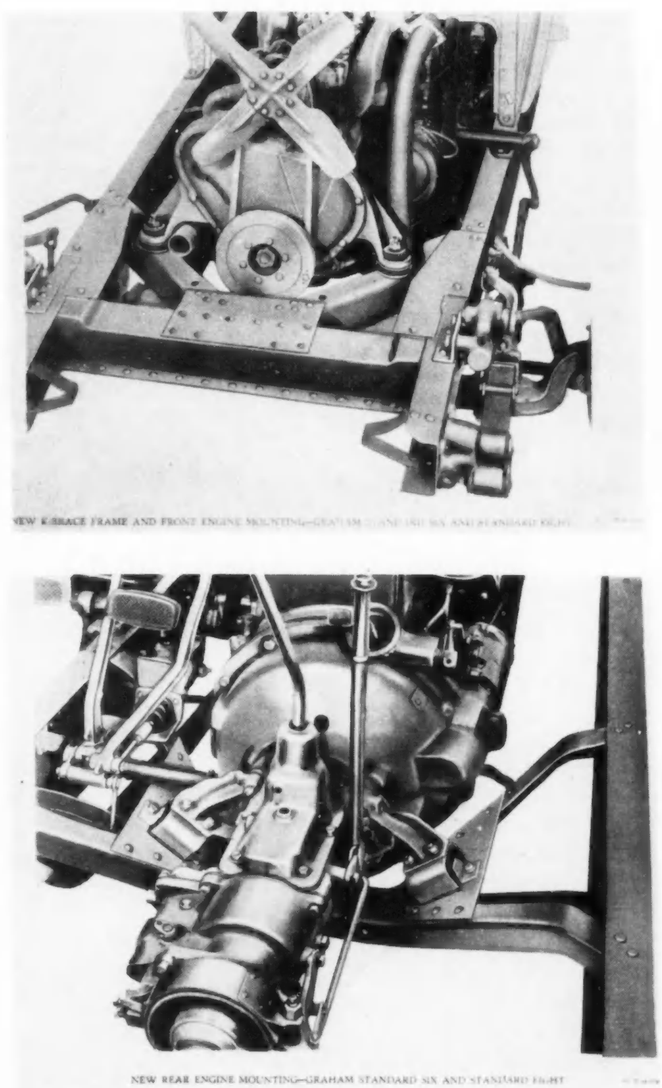
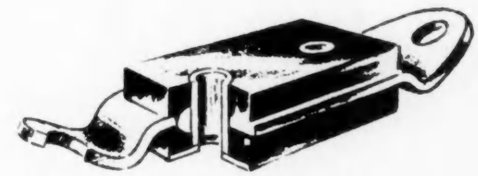


Fig. 13—Engine Mountings on the Graham-Paige Car

Master Chevrolet Side-Support

Designed to allow a universal motion in a plane parallel to the faces of the mount, controlled to close-rate limits, with a minimum deflection in a vertical plane, also controlled to close-rate limits.

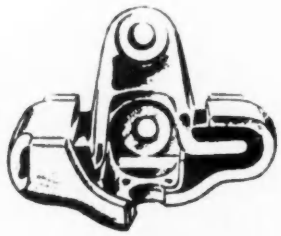
Construction: Two malleable-iron castings vulcanized in rubber.



Chevrolet and Pontiac Transmission Support

Designed to allow maximum side-deflection with no fore-and-aft movement, and small vertical movement.

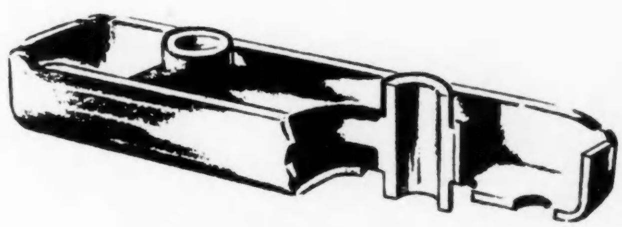
Construction: Rubber vulcanized to steel stampings. Tubular bushings inserted after molding.



Standard and Commercial Chevrolet Side-Support

Designed to cushion engine, replacing motor mounting bracket without change to adjacent members, allowing a restricted rocking motion and a minimum deflection in a vertical plane.

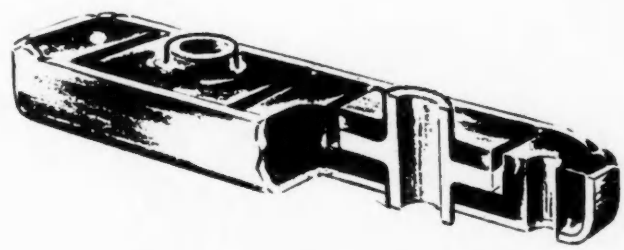
Construction: Rubber boots assembled to the legs of a steel forging, enclosed in a steel stamping to which a steel-rubber retainer is projection welded.



Chevrolet Passenger Front-Support

Designed to allow a definitely controlled swinging movement within close-rate limits, with restricted vertical deflection.

Construction: Malleable-iron casting and steel shell vulcanized in rubber.

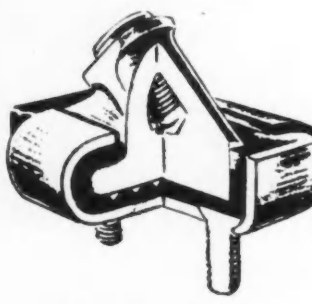


Chevrolet Commercial Front-Support

Designed to allow a definitely controlled swinging movement within close-rate limits, with restricted vertical deflection.

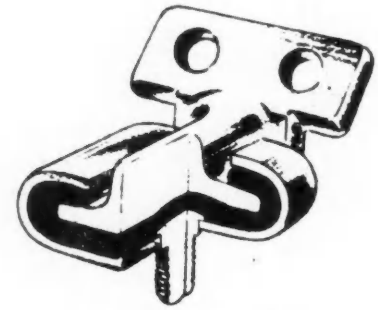
Construction: Two malleable-iron castings and steel shell vulcanized in rubber.

Fig. 14—A Series of Inland Mfg. Co. Mountings



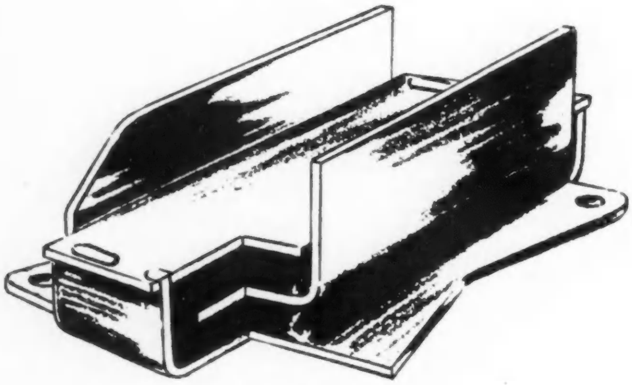
Olds Rear-Support

Designed to allow controlled close-limit deflections in three directions.
Construction: Rubber vulcanized to steel shell with bolts assembled. A malleable-iron casting is inserted after molding and the shell closed over.



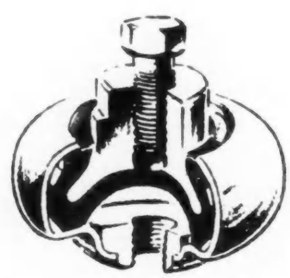
Pontiac Side-Support

Designed to allow controlled close-limit deflections in three directions.
Construction: Steel forging on steel shell with bolt inserted vulcanized in rubber. Shell is closed after molding.



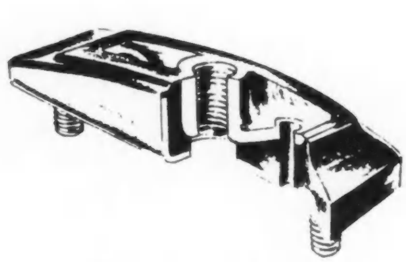
Buick Rear-Support

Designed to cushion motor with restricted motion in all planes.
Construction: Three steel stampings vulcanized in rubber. Outer housing and strap riveted after molding



Pontiac Front-Support

Designed to allow a restricted close-limit universal action in a horizontal plane with maximum vertical motion held to close limits.
Construction: Malleable-iron casting vulcanized in rubber and inserted in steel shell which is then closed over the rubber.



Olds Front-Support

Designed to allow a side-rocking movement with restricted fore-and-aft movement.
Construction: Malleable-iron casting and steel shell vulcanized in rubber. Bolts are inserted in steel shell before molding.

Fig. 15—A Series of Inland Mfg. Co. Mountings